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The Fan Manufacturers' Association

Guide to Fan Noise and Vibration

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PREFACE

This Guide has been prepared by the Fans Committee of the HEVAC Association and contributions to the text are gratefully acknowledged from the following members.

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GUIDE TO FAN NOISE AND VIBRATION

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INTRODUCTION

Whenever an air conditioning or ventilation system proves to be too noisy the immediate and instinctive reaction of the engineer, contractor and client is to blame the fan. This may well be understandable – but is it fair?

In most systems the fan is the original source of almost all the noise we hear; so it is all too easy to conclude that the fan is at fault and the blame thus fairly laid. However, in the hard world of contra-charges, 'extras' and liquidated and ascertainable damages, this conclusion inevitably carries the implication, not that the chosen fan is too noisy for the application, but that it is producing more noise than claimed in the fan specification itself!

Investigating why a system is too noisy for a particular application is like turning over a stone in the garden – except that in place of the conventional 'nasties' one finds ignorance, bad practice and prejudice – probably in larger measures than are found in any other area of engineering.

Helping to eliminate them is the aim of this Guide.

There are many excellent books on system noise and its control, but they start by assuming the fan sound spectrum is known. Unfortunately the fan sound spectrum, the starting point for our calculations, can itself be modified by the associated system. When this happens and the spectrum we start with is thus no longer the appropriate one for the particular system, what hope do we have of coming up with the correct answers at the end of our otherwise meticulous calculations?

This booklet then, apart from the first introductory chapter which is unashamedly taken from the Appendix of the successful HEVAC Fan Application Guide, confines itself to the acoustic and vibration characteristics of the fan itself – the starting points for meaningful systems analyses.

It is intended as a supplement to, not a substitute for, existing text books. Hopefully, it will enable engineers to avoid many of the problems now suffered unnecessarily.

1. NOISE

1.1. What is Noise

Most of the noise we hear is caused by fluctuations in the pressure of the air that surrounds us. It spreads out from its source in the form of pressure waves which vibrate our eardrums causing signals to be sent to the brain.

Because it is not economically possible to move air at reasonable rates without causing eddies, turbulence and pressure fluctuations (all of which produce noise), fans can be thought of as noise generating equipment. As a first stage, therefore, the engineer must know how noisy the fan is.

1.2. Sound Power & Sound Power Level

The noisiness of a fan could be expressed in terms of its Sound Power (the number of Watts of power it converts into noise). However, it is inconvenient to do this because the range of values found in practice would be incredibly large. (From 0.000 000 001 Watts to 40,000 000 Watts!) Fan noise is therefore expressed as a ratio which logarithmically compares its Sound Power with a reference power, the Picowatt (10^{-12} Watt). The unit of Sound Power Level is the Decibel.

Sound Power Level is defined as follows:

$$Lw = 10 \log_{10} \frac{W}{W_O} \text{ dB}$$

where

Lw = Sound Power Level in Decibels re 10^{-12} Watts.

W = Sound Power of the noise generating equipment in Watts.

W_O = reference sound power (re 10^{-12} Watts).

Fig 1.1 clearly shows how the logarithmic scale compresses the unacceptably wide range of possible Sound Power to Sound Power Levels having a practical range of 30-200dB.

Sound Power (Watts)	Sound Power Level dB	Source
25 to 40 000 000	195	Saturn rocket
100 000	170	Ram jet
10 000	160	Turbo jet engine 3200kg thrust
1 000	150	4 propeller airliner
100	140	
10	130	75 piece orchestra
1	120	Large chipping hammer
0.1	110	Blaring radio
0.01	100	Car on motorway
0.001	90	Axial ventilating fan (2500 m^3h) Voice shouting
0.0001	80	
0.00001	70	Voice — conversational level
0.000001	60	
0.0000001	50	
0.00000001	40	
0.000000001	30	Voice — very soft whisper

Fig 1.1.

A disadvantage of the Decibel scale is that values cannot be added or subtracted using the normal arithmetic rules. Decibel values must be converted back into absolute units of power (Watts), when they can be added or subtracted directly before re-converting back into Decibels. However, this tedious process can be avoided by using the following simple but approximate method. Column 2 shows how much must be added to the higher of two sound power levels to obtain the equivalent combined level, when the dB difference between the two levels is shown in column 1.

Column 1 Difference between the two levels dB	*Column 2* dB to be added to the higher level
0	3
1	2.5
2	2
3	2
4	1.5
5	1
6	1
7	1
8	0.5
9	0.5
10 or more	0

1.3. Frequency & Octave Bands

Noise usually consists of a mixture of notes of different frequencies, and because these different frequencies have different characteristics, a single Sound Power Level is not sufficient in itself to describe the intensity and quality of a noise.

The frequency scale is therefore split into bands and a Sound Power Level quoted for each. Octave bands are used for this purpose (bands of frequency in which the upper limit of frequency is twice that of the lower limit). If a more detailed picture is required one third octave bands are employed. The octave band frequencies universally recommended have mid-frequencies of 63, 125, 250, 500, 1000, 2000, 4000 and 8000 Hz.

The noisiness of a fan is described by a number of sound power levels (in Decibels re 10^{-12} Watts), each corresponding to an octave band frequency.

1.4. Sound Pressure & Sound Pressure Level

The Sound Power Level of a fan can be compared to the Power Output of a heater. Both measure the energy (in one case — noise energy, the other — heat energy) being fed into the environment surrounding them. However, neither the Sound Power Level or the Power Output will tell us the EFFECT on a human being in that surrounding space.

In the case of the heater, the engineer, by considering the volume of the surroundings, the characteristics of its envelope, and what other heat sources are present, can determine the resulting temperature at any point. In a similar way, the acoustic engineer, by considering very similar criteria, can calculate the Sound Pressure Level at any point. (Remember, it is sound pressure that impinges on our ears and determines how we hear noise.)

Sound Pressure Levels are also measured on a logarithmic scale. Again the unit is the Decibel, but this time referred to 2 x 10^{-5} Pa. (Fig 1.2) (Pa = N/m^2).

$$Lp = 20 \log 10 \frac{P}{P_O}$$

where

Lp = Sound Pressure Level in Decibels re 2 x 10^{-5} Pa

P = Sound Pressure of the noise in Pascals

P_O = reference pressure (= 2 x 10^{-5} Pa)

As is the case with Sound Power Levels, Sound Pressure Levels must be quoted for each Octave Band if a more detailed picture of the effect of the noise on the human ear is required.

There is another important benefit to be gained from using the decibel scale. Because the human ear is sensitive to noise in a logarithmic fashion, the DECIBEL scale more realistically represents how we respond to a noise.

Sound Pressure (Pascals)	Sound Pressure Level dB	Typical Environment
200.0	140	30m from military aircraft at take-off
63.0	130	Pneumatic chipping and riveting (operator's position)
20.0	120	Boilers shop (maximum levels)
6.3	110	Automatic punch press (operator's position)
2.0	100	Automatic lathe shop
0.63	90	Construction site — pneumatic drilling
0.2	80	Kerbside of busy street
0.063	70	Loud radio (in average domestic room)
0.02	60	Restaurant
0.0063	50	Conversational speech at 1m
0.002	40	Whispered conversation at 2m
0.00063	30	
0.0002	20	Background in TV recording studios
0.00002	0	Normal threshold of hearing

Fig 1.2.

THE ENGINEER MUST CLEARLY DISTINGUISH AND UNDERSTAND THE DIFFERENCE BETWEEN SOUND POWER LEVEL AND SOUND PRESSURE LEVEL. HE MUST ALSO APPRECIATE THAT dB RE 10^{-12} WATTS AND dB RE 2 x 10^{-5} Pa ARE DIFFERENT UNITS WITH NO ABSOLUTE FORMULAE CONNECTING THEM!

It is impossible to measure directly the Sound Pressure Level of a fan. However, the manufacturer can calculate this level after measuring the Sound Pressure Levels in each octave band with the fan working in an accepted standard acoustic test rig.

What he cannot do is unequivocally state what Sound Pressure Levels will result from the use of the fan. This can only be done if details of the way the fan is to be used, together with details of the environment it is serving, are known and a detailed acoustic analysis is carried out.

1.5. Specifying Sound Pressure Levels

Understanding the difference between Sound Power Level and Sound Pressure Level, the engineer must now learn to appreciate how acceptable levels of Sound Pressure can be specified.

As it is inconvenient to quote a series of sound values for each application, several efforts have been made to meaningfully express noise intensity and quality in one single number. Because the ear reacts differently to different frequencies all these single figure indices therefore mathematically weight the Sound Pressure Level values at each octave band according to how the ear responds to noise at that frequency.

1.6. dBA, dBB & dBC Sound Pressure Levels

What have been termed A, B and C noise levels were an early attempt to produce single number sound pressure indices. To obtain these, different values are subtracted from the Sound Pressure Levels in each of the frequency bands, subtracting most from those bands which affect the ear least. The resulting values are then added together logarithmically to produce an overall single number sound level. Fig. 1.3. shows the different weightings employed; they are known as the A, B and C curves. The resulting noise levels are known respectively as dBA, dBB and dBC.

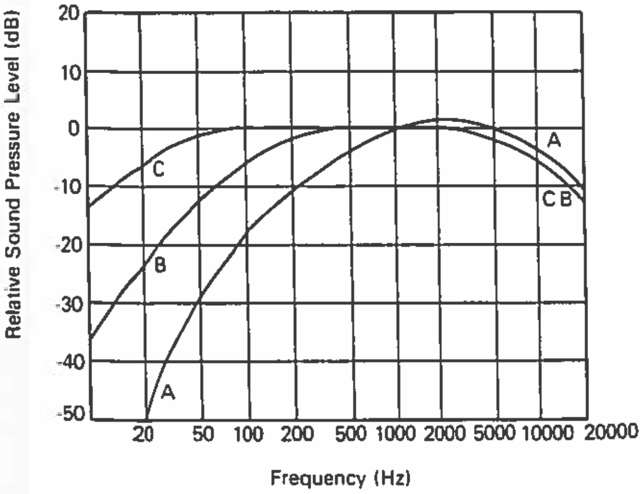


Fig. 1.3. Sound Weighting Curves

It was originally intended that the A weighting should be used for sound pressure levels up to 55 dB, the B weighting for sound pressure levels between 55 and 85, and the C weighting for higher sound pressure levels. However, nowadays, the A weighting is used for all sounds regardless of level because it has been found that there is a good agreement between subjective reaction and the A weighted sound level, regardless of level, for any generally similar noise sources.

A, B and C weightings are simple and effective for making initial assessments (inexpensive sound level meters are available which measure directly on these scales). Unfortunately, however, too much information is lost in combining all the data into one figure to allow them to be of any use for calculation and design work, and most noise control finally depends on frequency analysis.

1.7. Noise Criteria Curves (NC)

It is obvious that the combination of a single figure index such as dBA with more information on the shape of the frequency content would be useful. Noise Criteria curves (NC) were evolved to meet this need.

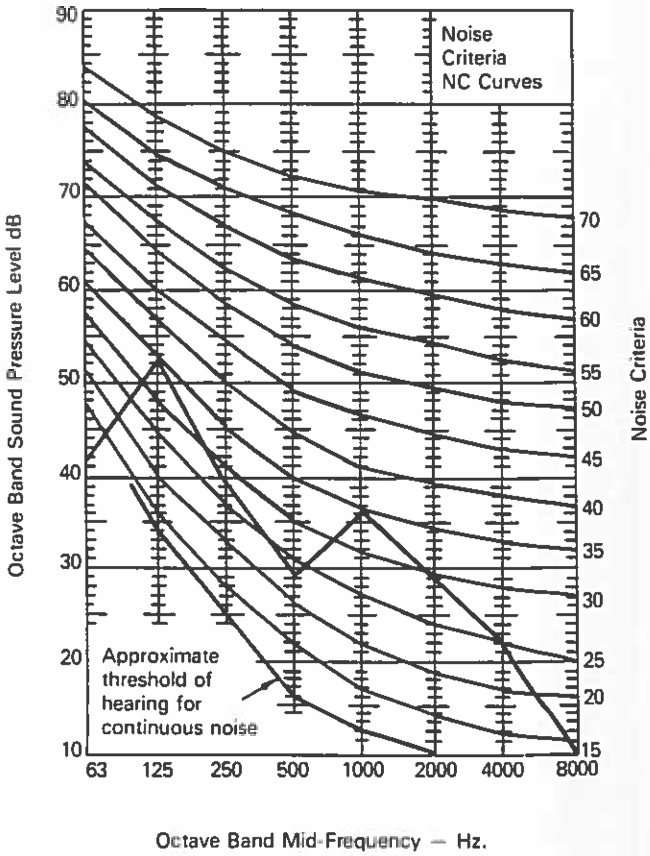


Fig. 1.4. Noise Criteria Curves

NC curves consist of a family of octave band spectra, with each curve marked with its own NC rating number. The octave band spectrum of the noise being analysed is plotted on the same grid and the NC rating of that noise corresponds to the highest NC curve touched by the noise spectrum.

Fig 1.4. shows a set of NC curves with overleaf a table indicating recommended levels for various environments. The spectrum of a noise with a NC rating of 35 is also shown on the grid.

NC ratings are particularly suitable for selecting and assessing suitable background noise levels for ventilating and air conditioning systems.

Recommended NC Levels for Various Environments

Environment	Range of NC levels likely to be acceptable
Factories (heavy engineering)	55-75
Factories (light engineering)	45-65
Kitchens	40-50
Swimming baths and sports areas	35-50
Department stores and shops	35-45
Restaurants, bars, cafeterias and canteens	35-45
Mechanised offices	40-50
General offices	35-45
Private offices, libraries, courtrooms and schoolrooms	30-35
Homes, bedrooms	25-35
Hospital wards and operating theatres	25-35
Cinemas	30-35
Theatres, assembly halls and churches	25-30
Concert and opera halls	20-25
Broadcasting and recording studios	15-20

1.8. Noise Rating Curves (NR)

NR (Noise Rating) curves are an alternative to NC curves. Because NR curves follow mathematical laws they are easier to handle on a computer than the empirically produced NC curves. The sets of curves are very similar and are used in an identical manner (Fig 1.5.).

NC and NR ratings are now being more frequently quoted on UK specifications and appear to be replacing the previously popular dBA level.

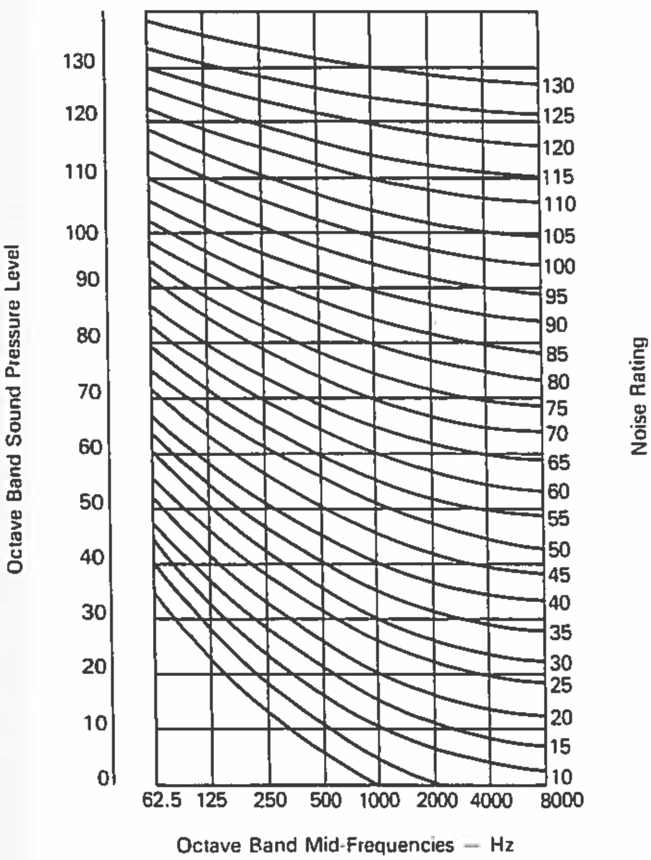


Fig. 1.5. Noise Rating Curves

Environment	NR criterion
Concert halls, opera halls, studios for sound reproduction, live theatres (500 seats)	20
Bedrooms in private homes, live theatres (500 seats), cathedrals and large churches, television studios, large conference and lecture rooms (50 people)	25
Living rooms in private homes, board rooms, top management offices, conference and lecture rooms (20-50 people), multi-purpose halls, churches (medium and small), libraries, bedrooms in hotels, etc., banqueting rooms, operating theatres, cinemas, hospital private rooms, large courtrooms.	30
Public rooms in hotels, etc., ballrooms, hospital open wards, middle management and small offices, small conference and lecture rooms (20 people), school classrooms, small courtrooms, museums, libraries, banking halls, small restaurants, cocktail bars, quality shops.	35
Toilets and washrooms, large open offices, drawing offices, reception areas (offices), halls, corridors, lobbies in hotels, hospitals, etc., laboratories, recreation rooms, post offices, large restaurants, bars and night clubs, department stores, shop gymnasis........................	40
Kitchens in hotels, hospitals, etc., laundry rooms, computer rooms, accounting machine rooms, cafeterias, canteens, supermarkets, swimming pools, covered garages in hotels, offices, etc., bowling alleys	45

NR50 and above
NR50 will generally be regarded as very noisy by sedentary workers, but most of the classifications listed under NR45 could just accept NR50. Higher noise levels than NR50 will be justified in certain manufacturing areas; such cases must be judged on their own merits.

WARNING

Unfortunately, a number of fan manufacturers indicate the noisiness of their products by Sound Pressure Levels expressed in dBA (or dBC). These figures refer to the Sound Pressure Levels which would be experienced by an observer at a distance of 3m, usually in dBA, or occasionally 3 x fan diameters (dBC) from the fan. It is important to note that these dB values must only be used to compare the noisiness of similar types of fan!

It cannot be stressed too often that on no account must the engineer be tempted to assume that the Sound Pressure Levels quoted in manufacturers' catalogues will in any way be similar to those achieved in practice.

Depending on circumstances, they can be substantially exceeded or reduced.

The internal areas of modern commercial and industrial buildings have hard boundary surfaces which cause a high proportion of sound energy incident upon them to be reflected causing a high reverberant sound pressure level to be built up. When this occurs, the sound pressure level readings indicated on a sound meter are independent of the distance from the noise source.

Thought of objectively, using dBA to describe the noisiness of a fan is as absurd as stating the output of a heater in terms of the temperature it would produce three metres from its face if it were suspended in space! It will be better for all involved when only Sound Power Level information is used to specify fan noise. In the meantime, engineers must always be careful to check how noise information is expressed.

In the same way, it is pointless specifying Sound Pressure Levels or NC ratings to a fan manufacturer unless adequate information about how and where the fans are to be used is also provided. In view of the considerable amount of work involved in calculating Sound Pressure Levels and NC criteria, engineers will be well advised to check whether a particular manufacturer includes this analysis work in his service or whether acoustic engineers must be employed.

1.9. Noise from Fan Systems

We can now assume that the engineer understands the fundamental difference between Sound Power Level (Lw) and Sound Pressure Level (Lp) and is aware of the main criteria by which acceptability is judged. Perhaps the theoretical calculation of the Sound Pressure Levels in an area resulting from a system feeding it may be beyond him (certainly the theory and method are beyond this brief treatise), but he should be in a position to understand the results of such an analysis. He must now learn what precautions he must take when designing and installing these systems.

The noise resulting from a fan system can be caused in several ways and can enter an area by more than one route.

A. The noise energy generated by a fan will be fed into the duct system, forced to pass along its length, and a proportion will enter the area being served by the system. The balance of original noise energy not reaching the ventilated area will be absorbed by the system.

B. Noise will also be generated by the airflow as it is obstructed and turned in its passage along the duct system. Sound energy will therefore be introduced into the system at bends, dampers, heater batteries, etc., and again a proportion will pass into the ventilated area. Also noise produced as the airflow passes through the terminal devices themselves will be radiated directly into the ventilated area.

C. Some of the noise absorbed by the duct system is in fact lost through the duct walls and can be a nuisance in those areas through which the ducting passes. This Break-out noise can even reach the ventilated area itself. Airborne noise from a plant room, and vibration energy from the fan, can also be transmitted through walls, floors and ceilings into adjacent areas.

1.10. Break-out & Flanking

Fig 1.6. shows how noise breaking-out of a system, or flanking it, can enter the ventilated area (and other areas).

(i) Airborne noise from plant room
(ii) Structureborne noise
(iii) Ductborne noise
(iv) Break-out of ductborne noise

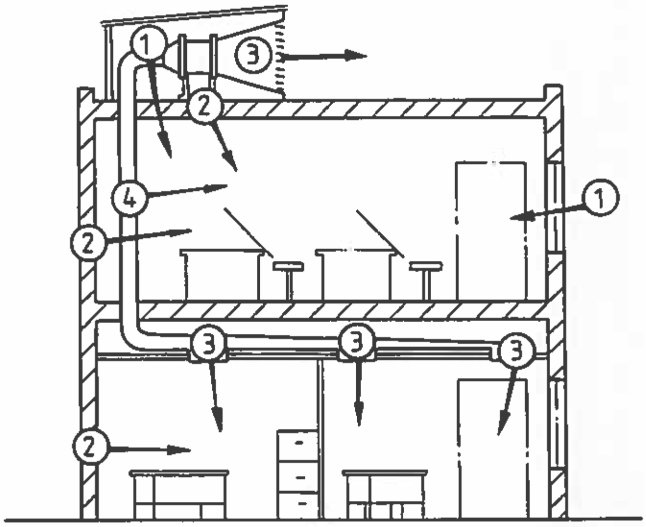


Fig. 1.6. Noise Breaking Out of a System

(i) **Airborne Noise** from the plant room can either be reduced by incorporating an attenuator after the fan itself, as shown in Fig 1.7, or if the plant room is noisy making the room as acoustically isolated as possible by using sound isolating materials and building methods.

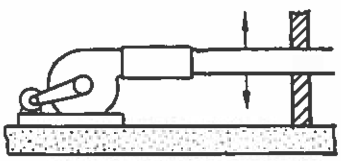
(ii) **Structureborne Noise** can be eliminated by isolating the fan from the building and duct system.

If a fan is belt-driven, it must be mounted on a frame together with its motor, and this common frame must then be supported on anti-vibration mountings.

All items held on this isolated frame must be reached through flexible connectors. For example, motor connections must be in flexible cable and flexible conduit. The flexible connectors between the fan and the inlet and outlet ducts must have similar sound reduction qualities to the ducts themselves, as otherwise noise will break-out through the connectors.

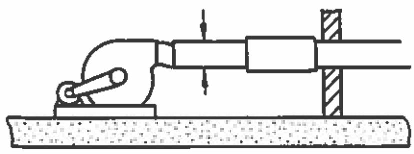
When the fan and motor frame is mounted on a firm base, anti-vibration isolators are easily selected from the manufacturers' catalogues providing the total weight of the frame (and what it is supporting) and the rotational speed of the fan are known. Nowadays, however, plant rooms are often at the top of a building and the deflections of the floor must be taken into account. The advice of the specialist isolator manufacturer should be sought in such cases. The isolation of vibration is an important part of Chapter 6.

PLANT ROOM WITH LOW NOISE LEVELS



Attenuator prevents duct noise from "breaking-out" into plant room.

PLANT ROOM WITH HIGH NOISE LEVELS



Attenuator prevents plant room noise "breaking-in" into duct system.

Fig. 1.7. Attenuator on Fan Discharge for Plant Room Considerations
See also Fig. 4.3.

It is possible for turbulent airflow to mechanically excite a large duct, but resilient duct hangers can prevent this vibration being transmitted to the building structure. Resilient hangers must be used immediately after the source of the turbulence (e.g., fan 90° bend) for a length of approximately 10 times the minimum duct dimension.

It should also be noted that turbulence induced vibration of ducts can result in significant sound radiation due to this phenomenon alone. To overcome this problem, first investigate the possibility of reducing the turbulence in the airflow within the system, possibly by the use of guide vanes, etc. If this is not practicable, then stiffen and/or damp the duct walls.

(iii) **Ductborne Noise** through the external louvre can be reduced by inserting an attenuator between it and the fan, or, if there is insufficient room, using an acoustic attenuating louvre.

(iv) **The Noise in the Duct,** and hence the break-out noise, can often be reduced by attenuators or occasionally with duct lining; but if this is not sufficient, outside lagging of the duct may be necessary.

WARNING
Noise, not produced by the ventilating system, can enter through the ductwork and subsequently be a nuisance in the ventilated area. This situation can be thought of as Break-in and must be treated in a similar manner to break-out. In the same way outside noise (e.g., aircraft noise) can enter the system through the external louvre and must be attenuated, if unacceptable.

1.11. Measuring Noise Levels

The acoustic performances of fan systems should be measured on site for the same reason that aerodynamic performances are checked; physical tests are the only certain method of ensuring that clients get what they specify!

The evaluation of a system on site is dealt with in Chapter 8. Examples of typical problems and their 'cures' are given in Chapter 10.

1.12. Summary

If the sound pressure levels exceed the specified values, the engineer must attempt to reduce them; all the time bearing in mind that time is short, no further money is usually available, and that it will be difficult, if not impossible, to gain access to certain sections of the system.

There are a number of actions which can be taken, but if none of these measures reduces the noise level sufficiently, then it may prove necessary to alter the acoustic characteristics of the area being served in an effort to absorb more of the sound energy discharging from the fan system. An acoustic engineer should be consulted if such a stage is reached.

However, everyone must be aware (especially clients!) that reducing noise levels below reasonable values is incredibly expensive. In fact, the total cost of a system increases exponentially as the NR rating is lowered. For this reason the target noise level resulting from a fan system must be chosen with care after studying the guides and the conditions special to the contract.

Remember — there is no justification in paying for a fan system which would generate a background level of only NR 25, if the noise from an adjacent road would make NR 35 more appropriate!

1.13. Fan Selection

Different types of fans produce different noise spectrum shapes and the engineer must make a selection that causes the minimum offence for the particular application. However, because certain sound frequencies are more easily suppressed than others, the choice of fan must be made after considering it alongside the absorption of the system itself, and often of a specialised attenuator device inserted into the system. For example, it can be advantageous to use a small high speed fan with a matched attenuator than a quieter slower speed fan which is more difficult to attenuate.

The subject of relative sound levels, spectrum shapes and attenuators is dealt with in detail in Chapters 3 and 4. The same chapters also discuss how fan noise varies with duty.

The engineer must also take care to ensure that the airflow into the fan impeller itself is uniform; otherwise the fan itself will produce considerably more noise than specified. Chapter 4 deals with the important aspect of system design.

1.14. System Design

The engineer should remember that noise generation within an air distribution system is caused by aerodynamic turbulence. If, therefore, he conforms to the codes of recommended design practice, paying special attention to those areas where the turbulence is likely, both aerodynamic and acoustic efficiencies will improve.

1.15. **Environmental Noise**

Often the external noise criteria can be more stringent than the internal criteria. It must be ensured that operation of fans does not affect local residents or any nearby properties, particularly if the fans are to be operated at night time.

The local Environmental Health Department may well be involved — they will place an overall noise limit to be met. This will usually be in the form of a maximum dBA level at a specified position. The value of the level is usually assessed with reference to the British Standard 4142, 1967 (revised 1975) "Method of Rating Industrial Noise Affecting Mixed Residential and Industrial Areas". The Standard compares the predicted noise level, correcting it for any tonal, impulsive and intermittancy and duration characteristics with the existing background level and concludes that if the corrected noise level exceeds the background noise level by 10 dBA then public complaints will be likely, whilst if the difference is 5 dBA then it is a marginal case. In general one usually designs to meet the existing noise level at the nearest property. Whether or not the fans are to run at night-time is thus crucially important in determining the degree of noise attenuation, as background levels usually fall considerably for this period.

2. FAN SOUND LEVELS (TESTING AND PRESENTATION)

2.1. **Testing Methods**

For the estimation of the noise radiated from a fan system, whether directly from an open fan inlet or outlet, or after passing through any ducting to which it is connected, an accurate determination of the fan sound level is necessary.

Various standard tests are prescribed to obtain the fan sound power level, using sufficient measurements of sound pressure level. As explained previously, the sound pressure level Lp is only an indication of the sound power output of a fan, and will vary widely according to the test environment. The fan sound power level L_W is calculated from one or more measurements of Lp at specified microphone positions together with the inclusion of factors related to the size and type of the environment. Four arrangements are used for standard tests, and are briefly described.

For full details, the test specifications should be consulted, the most relevant being:
BS 848 Part 2 Draft 1984 "Fan Noise Testing".
AMCA Standard 300-67 "Test Code for Sound Rating".

2.2. **In-Duct Measurement**

Measurements are made of sound pressure level inside the duct connected to the fan. Precautions need to be taken to avoid wind noise on the microphone and noise trapped in the duct by end reflections, and, therefore, microphone wind shields and anechoic horn type duct terminations are used. The in-duct method is very useful for testing the larger sizes of fan. Fig 2.1.

For this test, the relationship between L_W and Lp is given by $L_W = Lp + 10 \log A$ where A is the area of the test airway in square metres.

It must be said, however, that windshields offer only minimal protection against wind noise, and indeed aerodynamic nose cones and turbulence air screens on offer by noise measuring equipment manufacturers can give rise to high levels of wind-induced noise. Literature is usually supplied by the manufacturers relating such noise levels to air velocities. However, in practical situations large errors can arise. This type of test method must therefore be treated with extreme caution.

2.3. **Free Field Measurement**

As the name suggests, the open inlet or outlet of the fan is placed in an environment where the sound can radiate outwards in all directions without any reflections. Measurements are made of sound pressure level over the surface of an imaginary sphere, having its centre at the centre of the fan opening and a radius related to the opening size, but not less than 1 metre. Fig 2.2.

Hemispherical Free Field is a variant of this method where a reflecting ground plane is used and the measurements are made over a hemisphere centred on the ground vertically below the centre of the fan opening. This method is much more suitable for the testing of the larger fans because

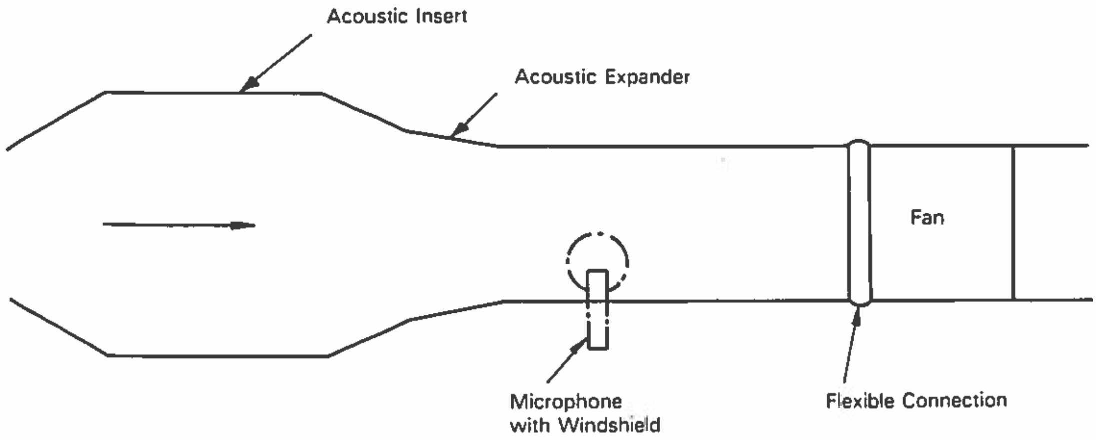


Fig. 2.1. In-duct Noise Measurements

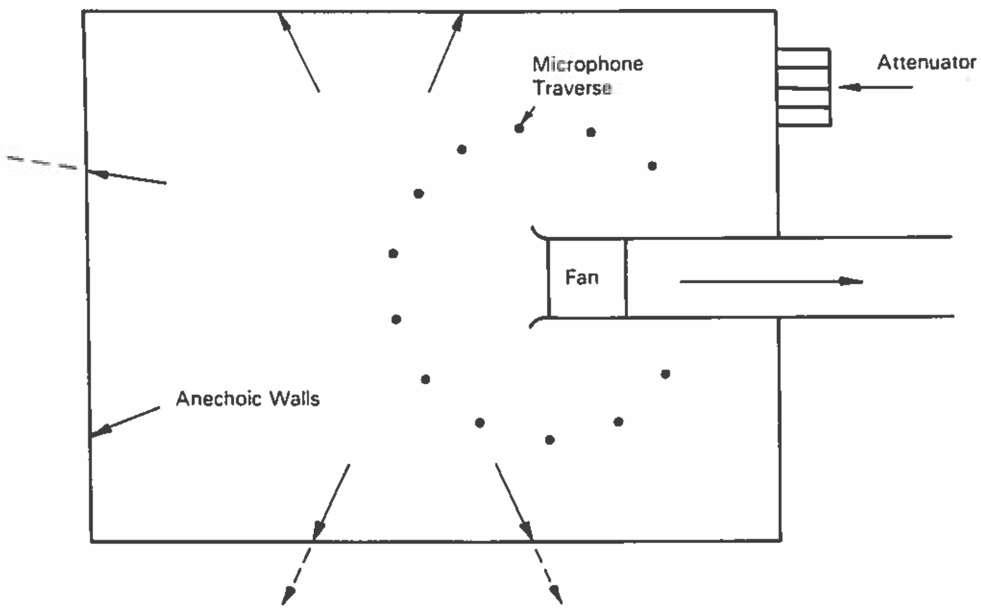


Fig. 2.2. Free Field Measurements

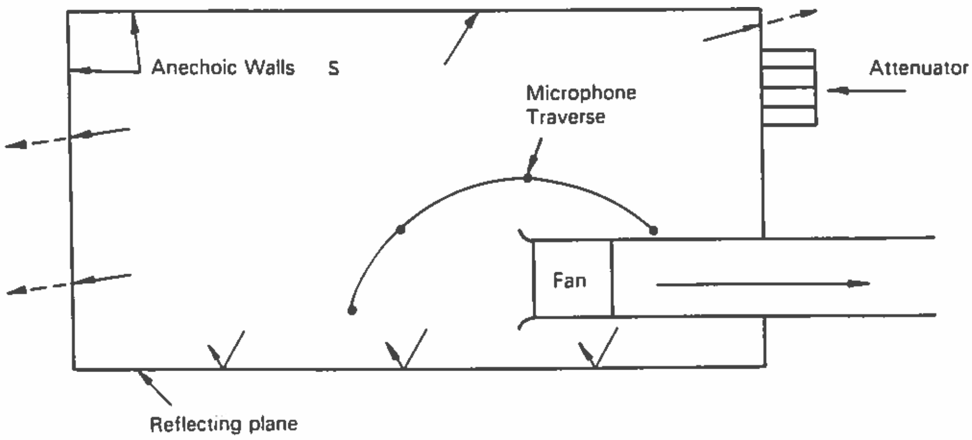


Fig. 2.3. Hemispherical Free Field Measurements

of the impracticability of obtaining free field conditions in the downward direction in those cases. It is particularly useful for testing out-of-doors, provided there are no adjacent buildings to reflect sound back to the measuring area. Fig 2.3.

For these tests the relationship between L_W and L_P is given by:

$L_W = L_P$ (mean + 20 log r + 11 — spherical free field
+8 — hemispherical free field

where r is the distance of the points of measure-ent from the centre of the sound source in metres.

2.4. Reverberant or Diffuse Field Measurement

The open end of the fan is situated in a room with reflecting walls and of such size and proportions that the sound is diffused into a uniform value within the room (except immediately adjacent to room surfaces). Measurements are made continuously over an arc in a number of distributed microphone positions to confirm that the sound pressure level is uniform. Sound power calculations are made using a room constant related to the size and room surface, or by comparing the sound pressure readings with those obtained from a reference source of known sound power output tested in the same way. Fig 2.4.

A semi-reverberant method is a variant which does not require the same control of room conditions, but uses the reference sound for a direct comparison of sound pressures and, therefore, power levels between the fan and a known source, subject to the same checks from the microphone readings to confirm that there are no undue variations. The method is particularly suitable for shop testing, provided that the background noise level is sufficiently low.

For these tests, the relationship between L_W and Lp is given by:

$$L_W = L_P \text{ (mean)} - L_P \text{ (r)} + L_W \text{ (r)}$$

where Lp(r) and L_W(r) refer to the mean sound pressure level and known sound power level of the reference sound source.

When a room fully complying with specification requirements for a diffuse field is used, the formula

$$L_W = L_P \text{ (mean)} - 10 \log T + 10 \log V - 14$$

may be applied where

T = reverberation time in seconds,
V = test room volume in m^3.

2.5. The Fan Installation for Test

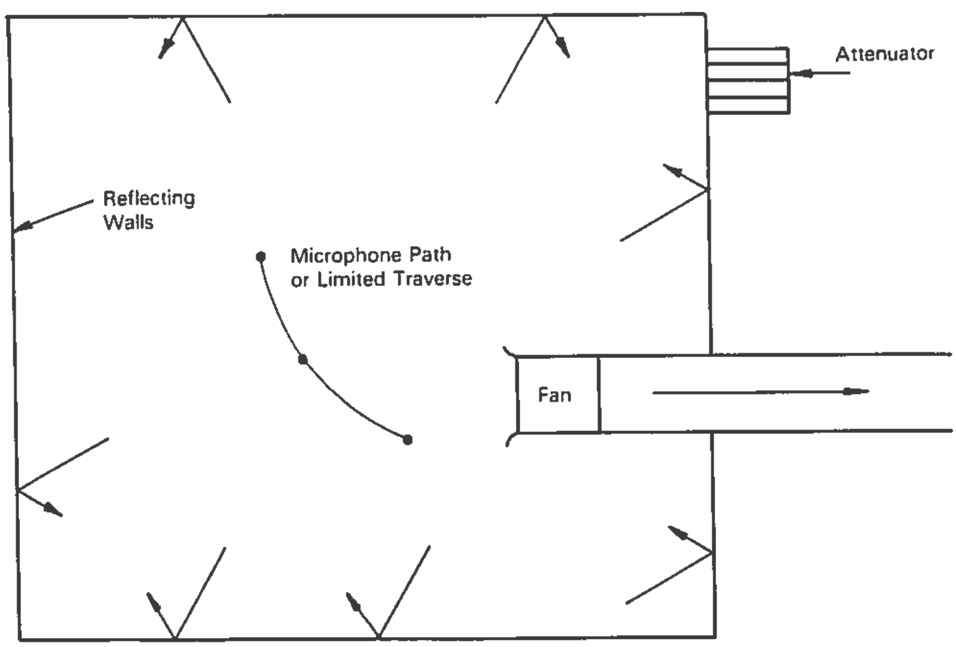
Whilst the standard test methods lay down the techniques for the accurate determination of the fan sound power level, it is also necessary in the fan installation to ensure that:

1. the fan is correctly loaded to its required operating point and that the manner of loading does not artificially increase the fan noise output by providing increased turbulence at the fan inlet, or by being a significant noise producer itself.

2. the sound issuing from the fan on the side opposite to that being measured is not creating an undesirable background level. This may be effected using acoustic attenuators.

   The background level in each octave band should be 10dB below the actual level of noise being measured.

3. Any unrepresentative noise and vibration coming from the fan and its driving means, or from its supports, are identified and eliminated before the test.



Diffuse Field — Test Method

Fig. 2.4.

2.6. **Site Tests**

The best environment for a site test is one which approaches one of the standard ones just described. This may be possible in certain limited cases, but in the main the situation is usually far from ideal, with high background levels. inaccessible measurement locations and intangible installation effects.

However, if the principles mentioned for the standard tests are used as a guide, it may be possible in some cases to obtain the fan sound level from a site test.

2.7. **Accuracy**

The typical uncertainty associated with the determination of fan sound power levels in a controlled test environment is indicated in the following table. A more detailed version is presented in BS 848 Part 2 Draft 1984. The uncertainties are greatest at low and high frequencies.

⅓ Octave Band Centre Frequency Hz	Typical Standard Deviation of Sound Power Level – dB
Less than 100	4
100 - 200	3
250 - 630	2
800 - 5000	1.5
6300 - 10000	3

When measurements are performed under on-site conditions the level of uncertainty is likely to increase.

2.8. **Presentation of Fan Sound Levels**

Because of the difficulty in accurately measuring sound power levels based upon sound pressure levels for frequencies lower than 100 Hz, the standard octave bands for which fan sound levels are presented are 125, 250, 500, 1000, 2000, 4000 and 8000 Hz. If 63 Hz values are reported, it must be appreciated that they will have a lower order of accuracy. Furthermore, if an overall sound power level is obtained from a single sound pressure level reading, using the unweighted or flat (dBLin) noise level meter circuit, it must be viewed with suspicion because of the possible inclusion of background low frequency elements of high magnitude.

Although in many cases the noise levels are not widely different, it should be realised that a particular fan could have four basically different sound outputs, according to the installation. They are:

Fan inlet sound level — ducted inlet
Fan inlet sound level — open inlet
Fan outlet sound level — ducted outlet
Fan outlet sound level — open outlet

See also BS 848 Part 2 Draft 1984.

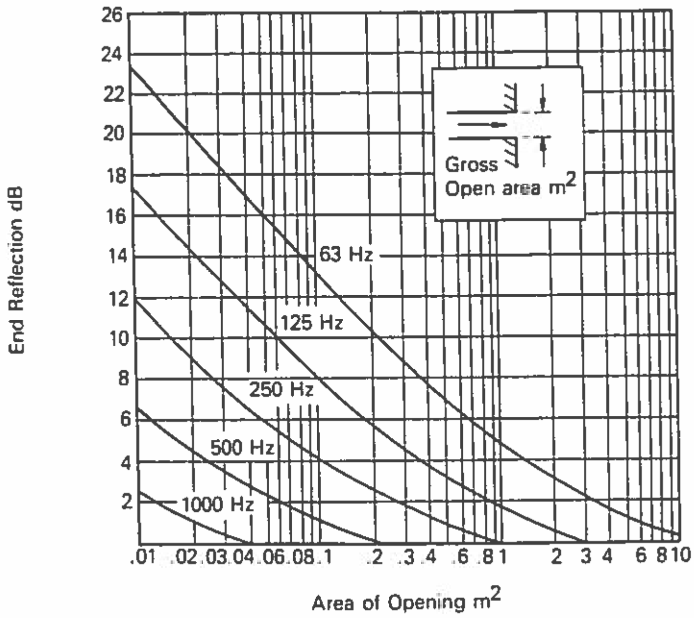
Only approximate relationships exist between these four, and when accurate noise calculations are necessary it is better to test the fan according to the desired installation (but refer to manufacturers' data).

However, in the absence of other data, it is often assumed that:

1. Fan inlet sound level (ducted inlet) = fan outlet sound level (ducted outlet).
2. Fan inlet sound level (ducted inlet) = fan inlet sound level (open inlet) + end reflection.
3. Fan outlet sound level (ducted outlet) = fan outlet sound level (open outlet) + end reflection.

The end reflection is related to the above octave band mid-frequency and the area of the opening, as shown in Fig 2.5.

The fan sound power level spectrum can be conveniently presented as an overall value, together with individual octave band level values relative to it (always negative).



End Reflection Correction

Fig. 2.5.

3. FAN SOUND LEVELS (OPERATIONAL SECTION)

3.1. Fan Selection

The choice of fan type is influenced by many factors, including sound. The fan sound spectrum is clearly a critical factor in relation to the achievement of a low noise level within the space served by the fan or adjacent to it. The overall sound level and in particular the shape of the spectrum are of major importance.

The type of fan, its size and speed will all influence the nature of the sound spectrum. Fig 3.1. shows some typical sound spectra for various types of fan.

However, excepting for fan coil units, system attenuation is usually strongly frequency dependent and therefore such a spectrum shape choice is not relevant.

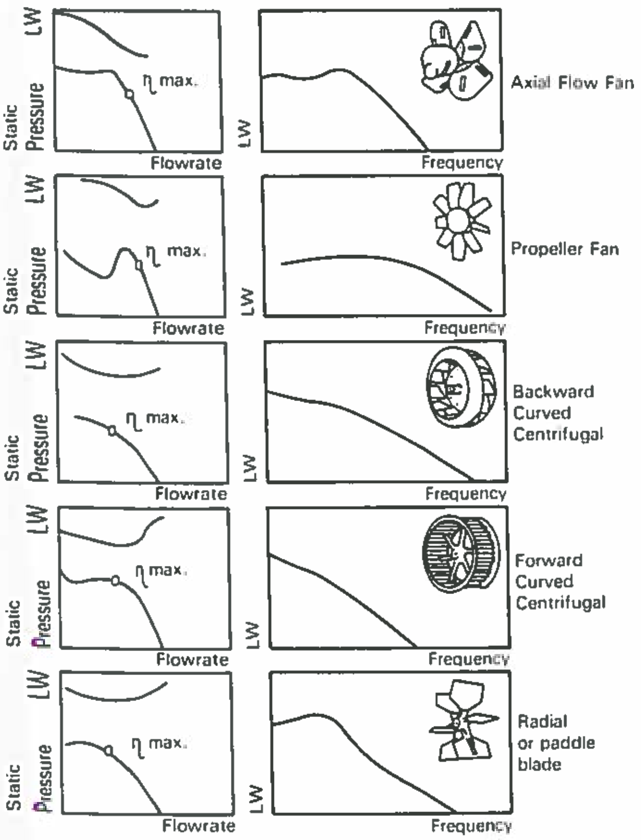


Fig. 3.1. Typical fan sound spectra for various types

For a given performance the lowest sound level for a particular fan type will generally be realized when the fan is selected to operate in the region of maximum efficiency.

3.2. Human Response

In considering the influence of spectrum shape, it is important to understand the response of the ear to sound. The sensitivity of the ear to sound varies with the sound frequency. It is less sensitive to low frequency sound and most sensitive to frequencies within the range 250 to 8K Hz. Fig 3.2. contains details of equal loudness contours. The influence of frequency on the ear's response can be clearly seen, e.g., at 500 Hz a Sound Pressure Level of 66 dB compares with a level of 80 dB at 63 Hz.

For a specified value the various noise criterion specifications such as NC and NR curves accept relatively high levels of low frequency sound, reducing as the frequency level increases.

In practice, this means that fan types with a spectrum shape which relates to the response of the ear will, as a basic fan, i.e. without attenuation, be generally more acceptable from a noise viewpoint than those types having the highest sound levels in the mid/high frequencies.

The general sound spectrum of centrifugal fans is normally highest in the low frequency levels, falling off as the frequency increases. They will thus appear quieter than axial types for a similar performance. It should be noted, however, that low frequency is generally more difficult to attenuate.

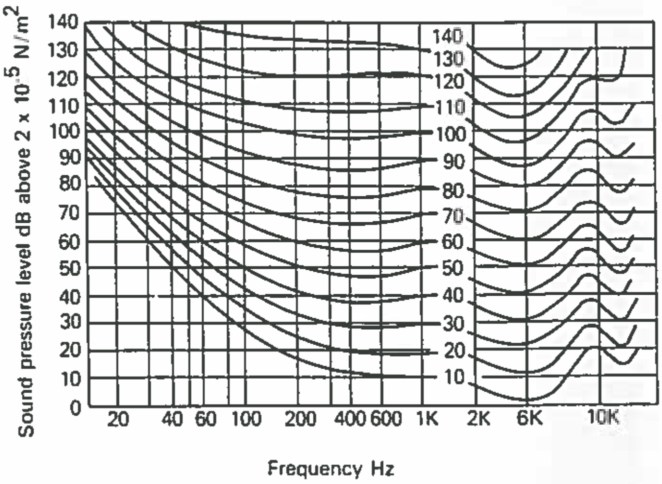


Fig. 3.2. Equal Loudness Contours (by Robinson and Dodson).

3.3. Tonal Effect

A further aspect that must be considered in relation to the quality of the fan sound is whether or not there are any tonal effects.

Fan sound is made up of two major components, broad band sound due to the aerodynamic noise generated by the throughput and the rotational sound attributable to the rotating parts at the blade passage frequency. The blade passage frequency and its harmonics is a direct function of the number of blades and the fan speed, i.e.,

$$\text{blade passage frequency} = \frac{\text{blade numbers} \times \text{rpm}}{60}$$

Certain types of fan can generate pure tones as part of the sound make-up and the pure tone can create a particular nuisance. Fans with a low number of blades, e.g., Propeller Fans and low solidity Axials, can generate pure tones and this possibility should be considered in the light of the application. Fig 3.3. shows ⅓ octave and spectrum of a 3 bladed Axial showing clearly the pure tone in the blade passage frequency.

In order to minimise the basic sound problem the fan selection must be carefully considered. The fan type and speed will influence the nature of the spectrum, whilst the operating point will affect the overall sound level.

It is emphasised that the presence of marked tones has a strong adverse subjective effect and it is the current opinion of many experts that our noise criteria and noise laws underestimate such effect.

3.4. Operation

The operating point of a fan can vary for diverse reasons and it is clearly important to be aware of the effect of a change in operation on the sound level.

Surprisingly, the variation in overall sound power in the region of normal operation is not excessive. In the case of centrifugal types the increase in overall sound power level for say ± 30% volume variation from the optimum selection point can be no more than 2 dB, depending on type.

In the case of Axial type fans the sound variation will again be a function of type and blade angle. However, at mid blade angles around peak efficiency some ± 20% volume variation can be required to bring about an increase in approximately 2 dB on the optimum point. At the steeper blade angles the overall sound level is remarkably consistent across the characteristic. At the shallow angles, as the peak pressure is approached the sound variation relative to change in volume increases and can be as much as 7 to 8 dB for a reduction in volume of some 20%.

Clearly this is a point worth noting since, providing the fan is selected as recommended, it is

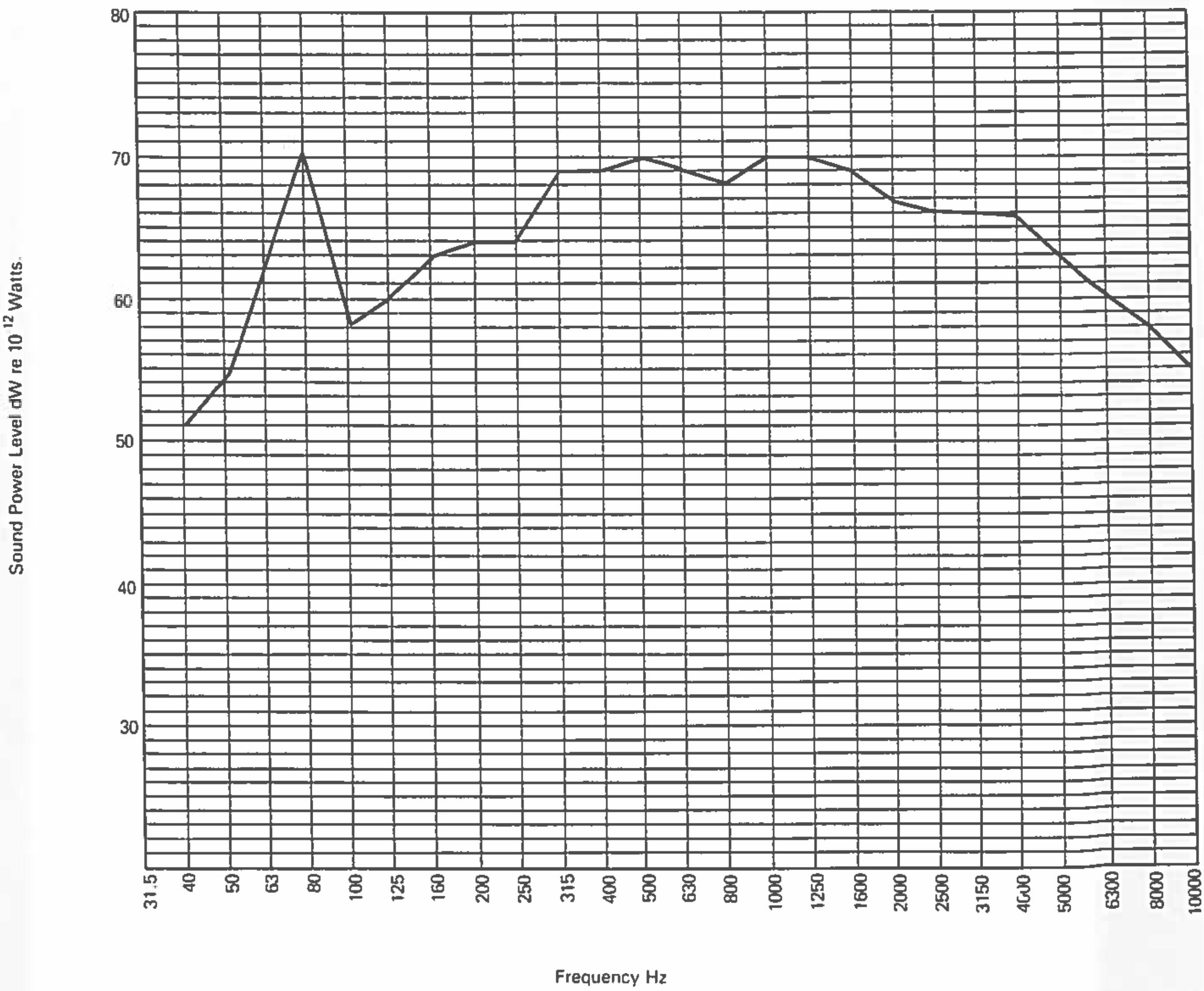


Fig. 3.3. ⅓ Octave sound power spectrum for 3 blade Axial at 1440 rpm showing pure tone effect.

possible to ensure that the effect on performance variation will not cause an excessive change in the sound level.

Care must be taken, however, in the case of a fan type having a pronounced stall. If an increase in pressure causes the fan to operate within the stalled region then the sound level will certainly increase and the performance will decrease. Though clearly not part of the recommended operating region, axial flow fans have a characteristic stall. Therefore, if there is the possibility of an increase in system resistance, the operating point must be carefully considered to ensure that a fan type having a stall will continue to operate within the recommended portion of its characteristic.

Generally, fan sound data are based on fan selection within the region of maximum efficiency. The operation of a fan at a point on the characteristic far removed from this region, say, shut off or free discharge, can bring about considerable increases in sound levels, of the order of 10 dB. The spectrum shape can also change.

Therefore, where, because of the application, fan operation is anticipated in these regions it is important to consult with the fan manufacturer for further advice.

3.5. Relative Fan Sound Levels for Different Fan Types

In comparing the differences in sound output of different types of fan, it is necessary to consider the basis on which they are compared, the installation arrangement being the same in each case, of course — i.e., open or ducted inlet or outlet.

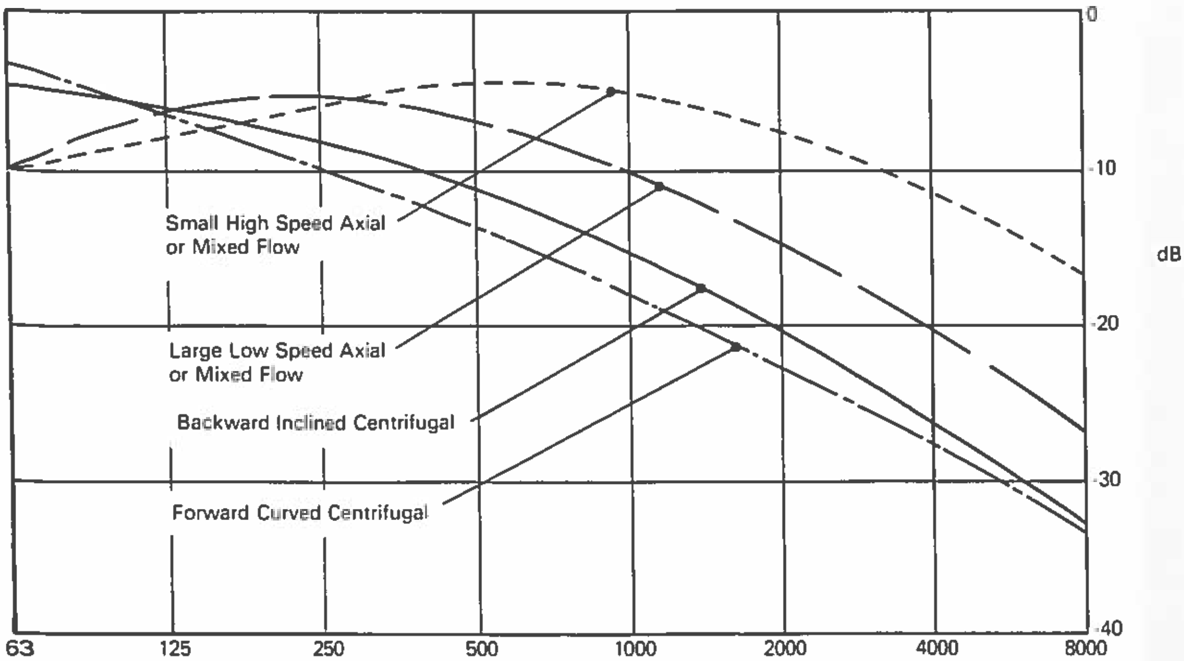
It is also necessary that the comparison is of fans operating at the same performance condition, and it will be readily recognised that a slow running centrifugal fan is likely to have a completely different sound output compared with that of a high speed axial flow fan.

1. **Overall Level**
   The first point to consider is the magnitude of the overall sound level. An approximate formula applicable to meet types of fan operating near their best efficiency point is:

   Sound Power Level $Lw(dB) = 33 + 9 \log_{10} P_R + 29 \operatorname{Log}_{10} \eta D$

   where P_R = impeller power (watts)
   η = rotational frequency (rev/s)
   D = impeller diameter (m)

   For a given performance requirement P_R will vary with fan efficiency and the product η D with fan type, being lowest with centrifugal fan types. It may be expected, therefore, that high efficiency centrifugal fans will be appreciably quieter than axial flow fans requiring a much higher blade tip speed. Typically, the ratio of tip speed is 2:1, giving a 9 dB difference in overall level.



Typical In-Duct Fan Sound Level Spectra
(Relative to oveall sound level which vaies with Fan Type for a specific duty)

Fig. 3.4.

2. **Spectrum Shape**
The second consideration is the shape of the sound level spectrum relative to that overall level. Fan sound level spectra have been fairly well established with typical examples shown in Figs 3.1 and 3.4.

It should be remembered that these figures for overall level and for spectrum shape are approximate and will vary from one fan design to another. For accurate data the fan manufacturer must be consulted.

3.6. Scaling Laws for Noise

It is possible to predict the change in noise level, both on an overall basis and on the spectrum values, with change of speed or diameter for fans of the same basic design. However, because of possible inaccuracies in the indices used in the scaling laws, they should not be used for changes in size, rotational speed, or tip speed greater than a factor of 1.5 greater or lower.

$$\text{Change in level due to speed change} = A \log_{10} \frac{n_2}{n_1}$$

$$\text{Change in level due to size change} = B \log_{10} \frac{D_2}{D_1}$$

Typical values of A are 50 to 55, and B are 70 to 75. Values of 55 and 70 respectively have been used in the nomogram, Fig 3.5, to permit easy calculation of the change in noise level ΔdB with change of speed and size.

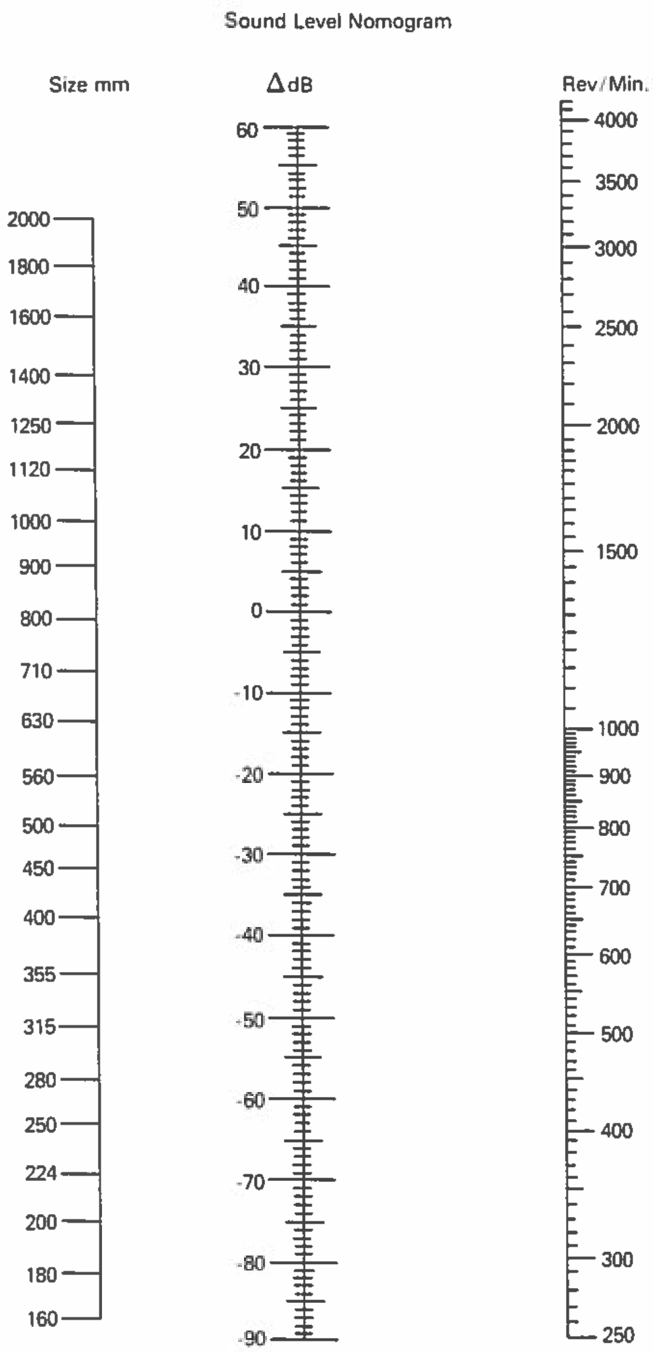


Fig. 3.5.

4. FAN APPLICATION AND INSTALLATION

4.1. **Fan Sound Level**

The sound emitted by a complete fan unit comprises:

a. Aerodynamic Sound resulting from the flow through the fan and casing.

b. Mechanical Sound, namely, the sound generated by the driving arrangement in the form of the bearings, any v-belts and the prime mover, generally an electric motor.

The aerodynamic sound is usually appreciably higher than the mechanical sound and, as such, the major cause of any nuisance value.

4.2. **Aerodynamic Sound**

The Aerodynamic Sound is a function of the fan duty, and this provides the starting point leading to the possible aerodynamic sound level. An alternative formula to that given in 3.5. can be used to estimate the Fan Sound Power Level.

$$L_W = 10 \log q_V + 20 \log p_F + K$$

where L_W = Fan Sound Power Level (dB re 10^{-12}W) at either the inlet or discharge side of the fan

q_V = Fan Inlet Volume

p_f = Fan Static Pressure

K = Constant, dependent on fan type, efficiency and operating point.

Clearly, the higher the fan duty, i.e., its volume and pressure, the higher will be the sound level and vice versa.

Therefore, the first positive step that the designer can take to minimise the noise level within the constraints of the type of plant or system is to ensure that:

(i) The volume to be handled by the fan is adequate, but does not include excessive contingency factors.

(ii) Careful consideration is given to the system layout and the air flow aspects of the component design to minimise the system resistance and hence the fan static pressure.

The volume is decided by process and/or application factors, and where reliable design data are available there can be little scope for reduction in the noise level. A key factor is the utilisation of reliable design data in the design computations to ensure that the estimated air quantity to be handled is sufficient to meet the process requirements.

The application of good aerodynamic design practice to the system layout will assist in minimising the flow resistance to air flow. Such factors as the choice of duct velocities, design of duct area changes and transitions, branch take-off arrangements, fan inlet and discharge connections, bend design and basic component selection relative to pressure drop must all be considered in the light of the plant layout restrictions, economics and function. There is clearly scope in this area for the skilful designer to influence at an early stage the basic fan sound level.

It is particularly important to ensure that all components are installed in such a manner as to achieve maximum efficiency. The air distribution must be as envisaged in the pressure drop calculations if the estimated values are to be realised.

It should be noted from the previous formula that the contribution from the volume factor is half that from the pressure factor. Hence a 20% reduction in the volume without change of pressure will reduce the overall Fan Sound Power value by 1 dBW, whilst a similar percentage reduction in the pressure with the same volume will lower the level by 2 dBW. There is often more scope through the application of good design practice to reduce the system pressure drop than the volume, and it is worth noting that the benefit in the overall reduction is greater.

Thus careful consideration of the plant and system factors influencing the initial Fan Duty, i.e., the fan volume and pressure, through the use of reliable design data and good system design from an airflow viewpoint, is the first step in minimising the primary sound level. Two additional benefits are a reduction in the power consumption and system regenerative noise.

TIME SPENT AT THIS EARLY STAGE COULD PROVIDE A THREEFOLD BENEFIT

Even when all these considerations have been taken into account, the fan aerodynamic noise can be significantly increased by allowing highly turbulent air and distorted flow to enter the fan. Care must be taken to avoid bends, dampers and obstructions near the fan inlet, which could give rise to these effects. On the fan outlet also, such obstructions should be avoided, particularly with a fan having a high outlet velocity. Fan performance also is affected (see HEVAC Fan Application Guide).

4.3. **Mechanical Sound**

The mechanical sound of a fan unit arises from its drive arrangement and includes any electrical hum from the driving motor. Where the motor is mounted in the airflow, such as a direct driven axial flow fan, its mechanical noise is additional to the aerodynamic sound carried in the ducting.

Fans which have an external drive arrangement will have a measure of sound from bearings, v-belts (where applicable), and the motor.

The level of the mechanical sound, certainly relative to the aerodynamic sound, does not normally reach a nuisance value. However, noise problems can sometimes result from the driving elements and consideration should be given to its possible reduction.

4.3.1. **Bearings**

Fan bearings are generally of the rolling type, incorporating ball or roller races. Good quality bearings do not usually give rise to a noise problem in relation to the overall fan sound level.

If roller bearings are defective due to wear or through surface irregularities within the rolling element, it will be apparent through an increase in noise level, which could in turn be unacceptable.

Care should be taken to follow the manufacturer's mounting instructions to avoid deformation of the rolling elements during installation. When non-pre-set bearings are used the recommended clearance setting between the rolling elements and the outer race is particularly important since bearing noise can be substantially influenced by this factor.

Sleeve or plain journals are generally quieter than ball or roller bearings, although normally the difference in relation to overall fan sound level is negligible. Sleeve bearings are generally used only on centrifugal fans when the noise requirements are extremely stringent, and to meet the particular load requirements of some types of large industrial fan.

Sleeve bearing noise level remains consistently low, whereas rolling bearing noise level increases with time as wear takes place and what is acceptable initially may not be so after a few years' operation.

4.3.2. V-Belts

Correctly selected, installed and maintained V-belts do not give rise to a noise problem. V-belt noise can, however, occur during start-up with a high pitched squeal arising from belt slip. The possible causes and corrective actions are as follows:

(i) Incorrect drive selection relative to the power to be transmitted and starting torque.
ACTION—Ensure V-belt drive is rated for total performance requirement.

(ii) Drive not re-tensioned after start up.
ACTION—Check and re-tension after initial running in period, say a few days, and thereafter in accordance with instructions.

(iii) In the case of Star Delta type motor control, speed changeover being at too low a speed through incorrect timer setting.
ACTION—Re-set timer in order that the speed changeover takes place nearer the maximum achievable speed in Star.

(iv) A motor having an inadequate starting torque will not provide sufficient acceleration and the "run-up" time will be extended. At changeover the speed can still be low relative to the full running speed and belt slip will take place when the Delta sequence is brought into operation.
ACTION—Change motor to a higher torque machine or fit fluid type or powder coupling.

(v) A further area of V-belt noise results from individual belt flap arising from the V-belts being of dissimilar lengths.
ACTION—Replace complete drive with a "matched set" of belts.

4.3.3. Electric Motors

Generally the main source of motor noise arises from the motor cooling fan and the passage of air through the machine. Cooling fan noise can become less predominant as the motor speed is reduced and, of course, is completely absent in the case of the non-ventilated motor. Other sources of noise can arise from bearings, residual unbalance and electro-magnetic effects (See Fig 4.1.).

Most motors use rolling type bearings and they can contribute to the noise level on very quiet installations. Generally, sleeve type bearings are only used on the smallest and largest machines or where the noise requirements are particularly stringent. When sleeve bearings are incorporated for additional quietness they increase the cost and, where V-belt drives are involved, introduce bearing load limitations.

Noise data are generally available from the motor manufacturer and it should be noted that they are related strictly to tests to the relevant Standard, viz., BS 4999 Part 51 : 1973. As in the case of fan sound data, the actual noise generated on site can be influenced enormously by the fan location. The data does, however, provide the basis for prediction, taking all relevant factors into account.

Drip proof enclosures are normally quieter than TEFC machines. Motor manufacturers can however supply what are termed "reduced noise" machines. These motors are de-rated to a maximum of 75/80% load within the chosen frame and a smaller cooling fan is fitted. An overall reduction of 8 to 10 dBA in noise rating can be achieved by this means relative to the standard industrial motor of the same rating.

Various types of acoustic enclosure can also be added to reduce the noise level. Relatively simple acoustic fan covers can reduce the level of the standard industrial machine of the same frame size by about 5 to 7 dBA (See Fig 4.2.).

Generally, single phase industrial motors are relatively noisy and their use should be avoided, if possible, where lower noise levels are required.

4.4. The Installation

Clearly, to minimise the effect of fan sound, the installation should comply with good practice.

It is particularly important to consider the siting of the fan in relation to critical areas from a noise viewpoint. This may seem obvious, but it is surprising the examples that can be listed where the largest highest speed fans are installed without due regard to their effect as a noise source. The case can be cited of a major University project where such a fan was immediately above the Bursar's office with the exhaust outlet above the office window!—the fan manufacturer was of course held responsible for the subsequent problems.

Clearly, "distances makes the fan seem softer" and, where possible, the further away the fan is from the more critical site areas the simpler and less costly will be the treatment to bring the sound to an acceptable level.

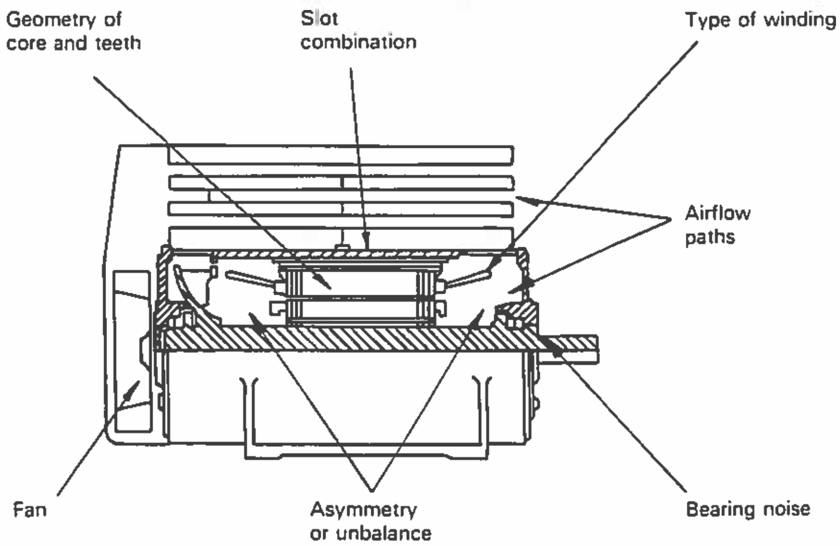


Fig. 4.1. Sources of Noise in an Induction Motor

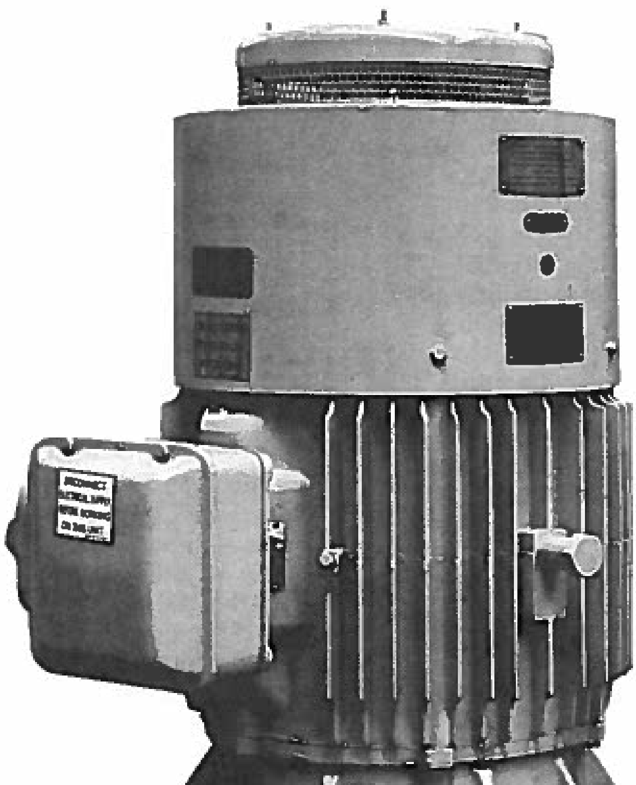


Fig. 4.2. Electric Motor fitted with Acoustic Fan Cowl

4.5. Noise Paths

From the sound viewpoint there are three aspects to consider:

1. **Ductborne Noise**
The sound transmitted along the ducting to the terminal points.
Using the inlet/outlet sound power spectrum as a basis, the sound radiated from the terminals of the ducting can be established. If necessary, duct attenuators can be installed to achieve the required sound level.

2. **Airborne Noise**
The sound radiated into the space around the fan installation. When the space around the fan is the problem area, the following must be considered:

(i) Sound radiated from the fan casing, which is influenced by the casing gauge and the physical size of the fan.
(ii) Sound break-out from the ducting and from flexible connections, again influenced by the material thickness and surface area.
(iii) Sound generated by the driving motors.
(iv) The reverberation effect of the plant room, where applicable, on the emitted sound level.

If the resultant sound level is higher than permissible, then one or more of the following features may be employed to reduce it:

(i) Insulation of the fan casing.
(ii) Insulation of the ducting and flexible connection (taking care not to rigidly short-circuit the flexible connections).
(iii) The fitting of a sound enclosure over the motor.
(iv) The fitting of a sound enclosure over the complete fan assembly.

3. **Structureborne Noise**
Structureborne noise can be substantially reduced by isolating the fan from the building and ducting. This is achieved by installing anti-vibration mountings under the complete fan set and fitting flexible connections between the fan and its ducting. Flexible connections are usually thin flexible material and therefore almost transparent to noise. Higher sound insulating connections are available, at the expense of some flexibility. However, they must always be regarded as a source of noise break-out.

If an accurate prediction of the sound level of the installed fan is to be made, it is essential that *all* the factors which affect it are considered.

Some of the above mentioned features are indicated in Fig 4.3.

Note: The positioning of the silencers adjacent to the Plant Room Walls and suitable lagging between Fan Discharge and Silencer Entry, thus reducing their direct proximity, can overcome the possible "bridging" of the silencer.

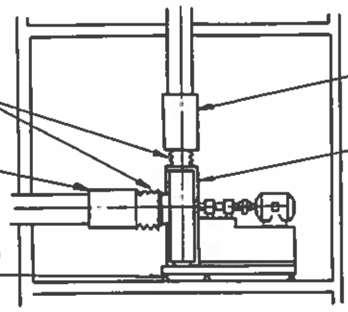


Fig. 4.3. An installed fan arrangement

4.6. Guaranteed Systems

In the case of industrial type fans handling material or corrosive gases and where reliability is of prime importance, the fitting of a vibration transducer to the bearings for condition monitoring is often recommended. Such a device will indicate the increase in the level of vibration and enable the plant operator to plan the necessary action to correct the situation. This is clearly preferable to the unexpected catastrophic breakdown which can sometimes result from either a bearing failure or an undetected impeller failure through excessive material build-up, resulting in imbalance and a high increase in vibration level. See Chapter 7.

It is important to realistically relate the specified sound levels to the environment. Instances have occurred where sound levels have been specified for fan equipment which have been considerably less than the actual plant background level. Clearly, the cost of the installation to the client was therefore considerably higher than necessary.

Silencing can be expensive and can be more costly than the basic fan and motor assembly.

4.7. Airstream Devices

It is important to consider the influence on the fan sound of the various airstream devices which can affect the fan sound level.

4.7.1. Flexible Connections

Flexible connections are often fitted to the fan inlet and discharge. It is essential to ensure that they are correctly fitted, i.e., reasonably taut while not rigid and not off-set in relation to the fan flanges or spigots. Incorrectly fitted flexible connections can cause an increase in sound levels. In the case of high pressure exhaust or supply fans, flexible connections should be arranged with a ducting sleeve arrangement to cater for the pressures or suctions involved. Off-set connections or bends close to the fan inlet can cause asymmetric inlet flow which can substantially affect the sound levels.

4.7.2. Inlet Cones

Open inlet fans should be fitted with inlet cones to ensure good inlet flow conditions and the fan should be sited at least 0.75D (where D is the impeller diameter) from facing adjacent walls or enclosures if the inlet flow and thus performance is not to be appreciably affected.

4.7.3. **Dampers**

Multi leaf type dampers close to the inlet can cause substantial regenerative noise and also affect the flow and thus the sound level. If the fan performance is too high in the case of, say, a V-belt driven unit, rather than use a damper, it is much better to correct the volume by reducing the speed. The kW is proportionate to $Speed^3$ and the Lw proportionate to $Speed^{5.5}$. Thus the sound level can be reduced, energy saved and cost of adjustment recovered quickly.

4.7.4. **Guide Vanes**

Inlet guide vanes are often fitted to centrifugal fans to attain efficient performance variation. It is important to note that such devices can increase the fan sound level. The effect on the fan will depend on fan type and the operating point, but a 3 dB increase can be brought about by a volume reduction of around 25%.

4.7.5. **Bearing Position**

To save space, centrifugal type fans can have the impeller mounted between bearings, with a bearing sited in the fan inlet. Normally, however, this arrangement, on a correctly selected fan, does not have an appreciable effect on the sound level.

4.7.6. **Motor Position**

In the case of Axial Flow fans, it is usual to operate the fan with the motor positioned downstream. The fan sound level is sometimes increased with the motor in the upstream arrangement, but any change is very much a function of the details of the motor mounting. Fig 4.4. shows comparative test figures for a typical 9 bladed Axial with the motor in the alternative positions.

	OCTAVE BAND MID FREQUENCY							
Motor position	125	250	500	1K	2K	4K	8K	Hz
Downstream	72	80	82	79	79	74	70	dB
Upstream	79	90	86	84	81	74	69	dB

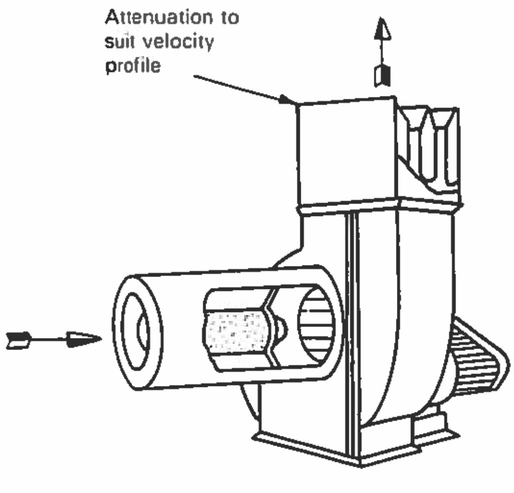
Fig 4.4. Axial Flow fan Sound Power values with alternative motor positions. Fan operating duties similar.

4.7.7. **Attenuators**

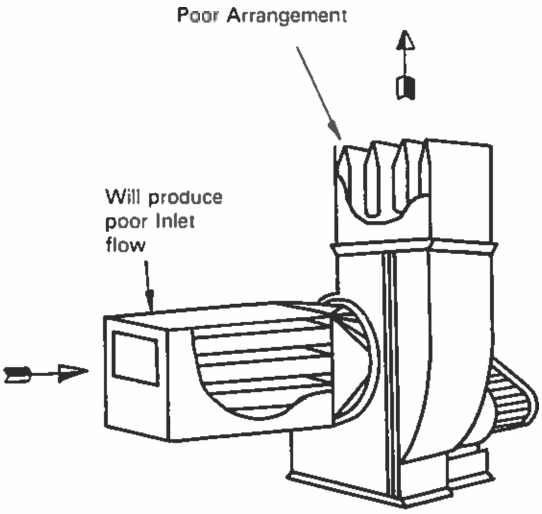
The arrangement of attenuators must be considered carefully and Fig 4.5. shows preferred arrangements. It is important to arrange the discharge attenuator splitter correctly in relation to the fan outlet velocity profile in the case of a centrifugal type. This point also applies to the arrangement of discharge multi leaf dampers. It is recommended that the discharge attenuator on a centrifugal type fan incorporates a short plenum prior to the splitters in order to assist in equalising the velocity profile through the attenuator and thus ensure the performance complies with design requirements.

Where cylindrical attenuators employing a centrebody are fitted to an axial flow fan, it is important that the diameter of the centrebody closely matches that of the impeller hub, especially if fitted on the inlet side. An incorrectly matched centrebody can cause the fan to generate more additional noise than the attenuator absorbs.

Clearly, careful consideration must be given to the fan inlet and discharge arrangements to ensure a good sound performance basis.



Attenuator Arrangement (Recommended)



Attenuator Arrangement (Not Recommended)

Fig. 4.5. Preferred Attenuator arrangements — Centrifugal fan

5. SILENCING OF FAN NOISE

5.1. Sources of Noise

Noise from a fan or fan system is transmitted to a listener along three principal transmission paths — the air, the duct, and the structure. Although all noise is ultimately airborne to the ear of the listener, the term airborne noise, though strictly inaccurate, is generally applied to noise from the open inlet or outlet of the fan, noise radiated from the fan casing and noise from the fan drive. Airborne noise can create noise problems in the immediate vicinity of a fan, e.g., in a plantroom, or remote from a fan installation, for example in the neighbourhood environment.

Ductborne noise, as its name suggests, travels along the ducting attached to the fan, and may reach the listener either by being radiated from the end of the duct or by breaking through the walls of the duct. Ductborne sound may be generated directly at the fan, but it can also include noise generated by the flow of air over elements of the ventilation system such as dampers, regulators, etc. Fan air ducts, especially ducts of circular cross section which have relatively rigid walls, are extremely good at transmitting noise without significant energy loss over long distances. Large rectangular ducts of welded sheet construction have relatively flexible walls and there is some attenuation of low-frequency, in-duct noise, much of the lost energy being radiated out through the duct walls and potentially causing a further problem.

In a 1-metre square duct the in-duct noise level attenuation at 125 Hz is initially of the order of 1 dB per metre, while at 500 Hz it is less than 0.3 dB per metre. Radiussed right-angled bends can give reductions of up to 3 dB at high frequencies while mitred bends can give 6-8 dB at certain mid-to-low frequencies, although, again, this is at the expense of increased breakout noise levels.

Structure-borne noise can be the source of major noise problems at great distances from the source. Vibrational energy from the source is coupled directly into the building structure normally only if there is good physical contact between the source and the structure. This can be prevented by the use of appropriate antivibration mountings and Chapter 7 discusses this aspect in some detail.

Control of airborne and ductborne noise can be achieved by the application of a number of techniques. The use of silencers (sometimes referred to as attenuators or mufflers) attached close to the fan or installed in the duct work can reduce airborne noise from an open fan inlet or outlet and the ductborne noise. The application of lagging treatment can control noise from the fan casing and noise being radiated from ductwork. The provision of enclosures can control the propagation of noise from motor drives and the fan installation in general.

This chapter deals with some aspects of the design performance and use of silencers, acoustic lagging and enclosures.

5.2. Duct Attenuators.

5.2.1. Basic Types

Sound attenuators may be divided into three classifications — active, reactive, and resistive. Active type silencers operate by positioning loudspeakers in the vicinity of the source and driving them with noise signals which cancel out the source signals. They are still very much at the development stage and are unlikely to be used in general ventilation applications in the near future.

Reactive types of silencer consist of an expansion chamber or chambers with interconnecting pipes. They operate on the principle that when a sound wave in a duct arrives at a discontinuity only part of the sound energy is transmitted through the discontinuity, the remainder being reflected back towards the source. The transmission of sound is reduced even although the discontinuities may not absorb any sound energy.

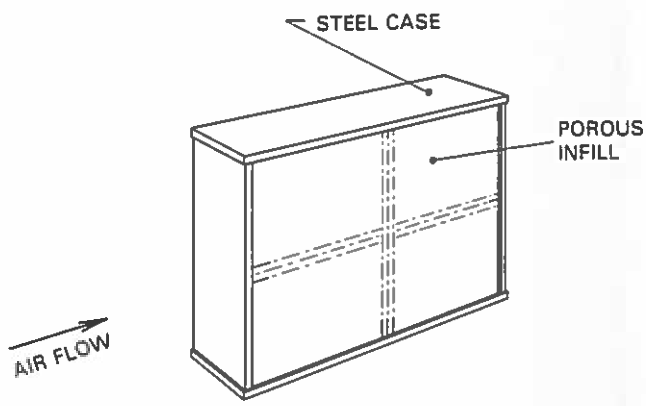


Fig. 5.1. Splitter Construction

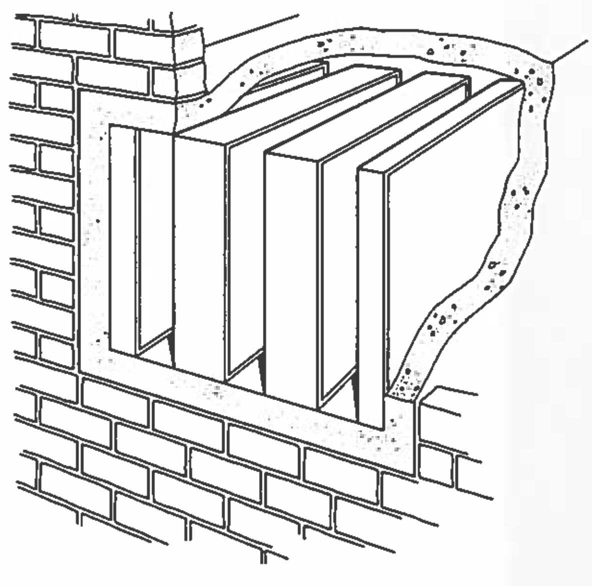


Fig. 5.2. Typical Splitter Installation

In contrast, resistive silencers, which consist of a porous or fibrous material placed in the flow as indicated in Figs 5.1. and 5.2, absorb energy directly. The incident sound waves cause the air in the pores or between the fibres to vibrate and may also cause the skeleton to vibrate. Energy is absorbed due to air friction in the pores, internal friction in the compliant skeleton and heat loss from the compressed air to the solid skeleton.

Purely reactive silencers are not widely used in fan applications for several reasons. They are generally highly tuned, providing substantial attenuation only over a relatively narrow range of frequencies. They are usually most effective at low frequencies, but tend to be bulky and have high aerodynamic pressure losses. Reactive silencers are used in certain special circumstances, for example, in silencing reciprocating compressors and in silencing contaminated airflows where the contaminant would react with or block the porous lining materials of a resistive silencer.

There are many types of resistive or dissipative silencers, but in the forms most often encountered consist of sections of rectangular ductwork containing splitters of acoustically absorbent infill (Figs 5.3. & 5.4.) or circular section units with acoustic absorbing material on the internal walls (Fig 5.5a.). Circular type attenuators are often fitted with an absorbent lined pod concentric with the duct axis (Fig 5.5b.). Other fairly common silencer elements include lined bends (Fig 5.7.) which provide a greater attenuation than an equivalent straight unit with the same volume of absorbent material (Fig 5.6.).

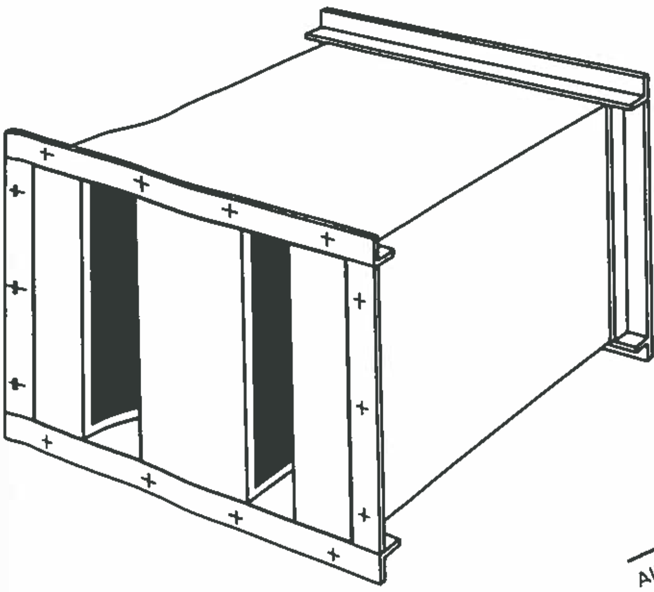


Fig. 5.3. Splitter Attenuator

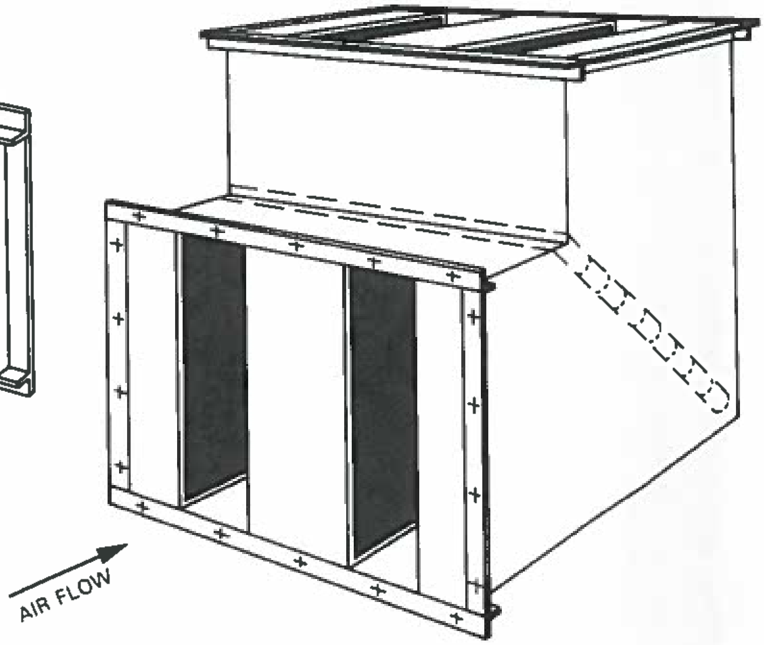


Fig. 5.4. Bend Splitter Attenuator

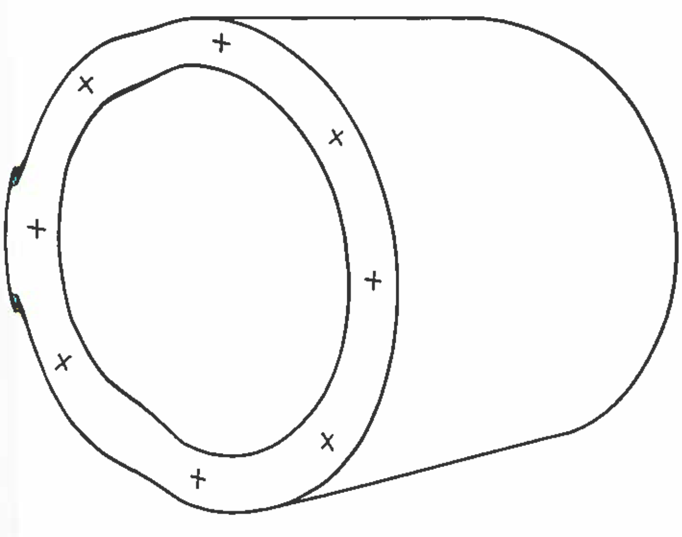


Fig. 5.5a. Cylindrical Attenuator

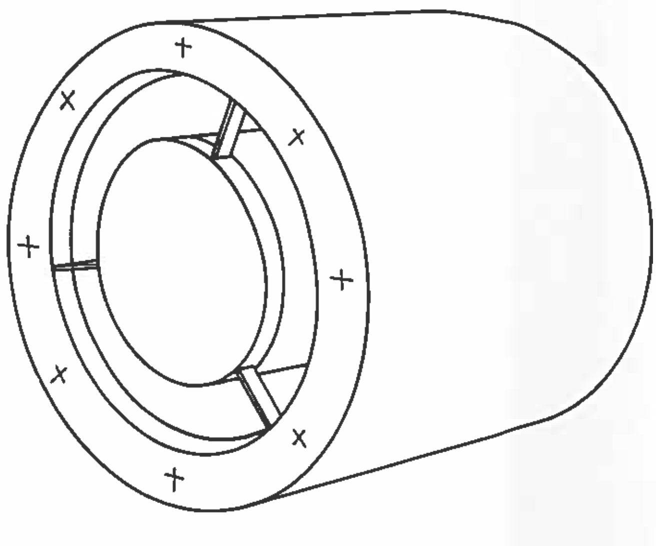


Fig. 5.5b. Cylindrical Attenuator with Centrebody

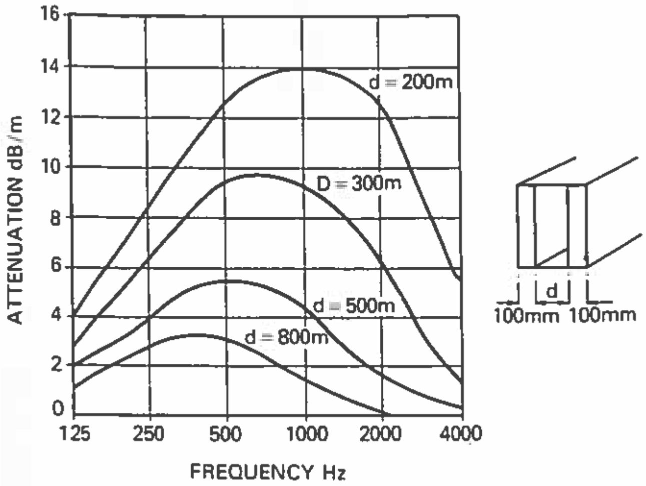


Fig. 5.6. Attenuation in ducts lined with 100mm thick rockwool blanket nominal density 80kg/m^3 and airspace d.

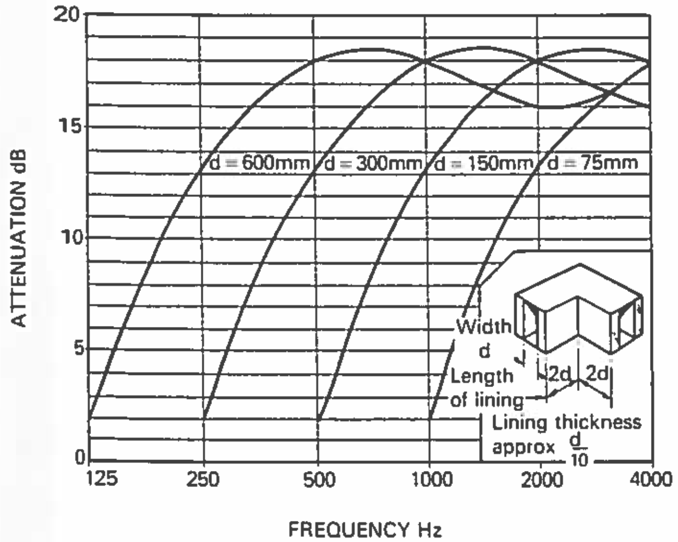


Fig. 5.7. Attenuation from lined bends.

5.2.2. Materials and Construction of Resistive Silencers

The acoustic lining material is obviously of paramount importance in determining the performance of a silencer. The selection of a material depends not only on its acoustic performance but a number of other specific design criteria including flammability, mechanical strength, resistance to erosion from high gas flow velocities, resistance to contamination from water, oil, dust, etc. The most commonly used materials are glassfibre and mineral wools. Some foamed glasses or metals are occasionally used. Foamed plastic materials are, in general, to be avoided as potential fire and smoke hazards. The fibrous materials are sometimes lightly resin-impregnated to form slabs or faced with a lightweight open mesh scrim or cloth. Non-porous membranes are used in cases where contaminants are present in the air stream or where precautions must be taken to avoid contamination of the airflow from particulate matter in the silencer, for example, in clean rooms or in hospital operating theatres. The acoustic properties of the absorbing material can be strongly affected by such membrane material unless it is very thin (e.g., 30 micron), in which case problems of mechanical strength arise. To retain the non-porous membrane, perforated metal sheet with about 40 per cent per open area is often placed in front of the lining. This perforated facing can improve the low frequency attenuation characteristics of the silencer. The casing of the silencer should be of a construction at least as robust as the ducting to which it is connected. Further information is given in the section headed duct breakout noise.

5.2.3. Noise Attenuation Characteristics

Resistive silencers consist basically of a duct with some acoustic lining. In a treated duct, for a given sound pressure level, the absorption of energy is proportional to the area of lined surface, while the amount of sound energy is proportional to the cross-sectional area of the duct. The energy removed will vary directly with the ratio of lined perimeter to cross section area, indicating that narrow ducts, which have high perimeter : area ratios, are acoustically preferable to wide ducts (Fig 5.6.).

For frequencies for which the duct width is greater than a wavelength, the sound field begins to recede from the walls as the sound energy is removed by its interaction with the porous lining and the energy is concentrated towards the centre of the duct. In any silencer, the mid and high frequency attenuation is improved by having lined splitters or pods to reduce the passage width. Their installation, however, increases the aerodynamic pressure loss.

For any lining material, the low frequency attenuation increases as the lining thickness is increased, and for good low frequency attenuation thick linings are necessary.

Sound energy travels down a duct as a series of pressure waves. Some of the waves travel with their wave fronts progressing axially along the duct. In a rectangular duct other waves travel by reflecting off the duct walls so that the wave fronts effectively zig-zag down the duct. In circular ducts the wave fronts spiral down the duct. These patterns are described as modes and the higher the mode the more complex is the path of the wave front along the duct. At high frequencies a large number of modes can exist, while below a certain frequency, which is a function of duct diameters, only one mode, the plain wave mode, is present. In a lined duct the energy in the higher order modes is absorbed more rapidly than that in the low orders so that, initially, attenuation may be quite high.

As sound energy is initially absorbed there is a redistribution of the energy towards the centre of the passageway and the rate of attenuation with distance decreases. Increasing the length of a silencer will increase the attenuation; however, the increase is not linear.

If a number of silencer units are required and their combined performance has not been experimentally assessed an allowance must be made for this performance shortfall. On some occasions, if single units were installed about 10 diameters apart the overall attenuation would be closer to the sum of their individual attenuations.

5.2.4. Effect of Airflow

Airflow through a silencer has at least two effects — it can alter the attenuation from the level measured with no airflow, while the flow of air through the passageway can itself generate noise. An approximate expression for the change of attenuation with flowrate is

> Attenuation with flow = no flow attenuation x $(1 \pm M)$ where M is the airflow Mach number and the + and − signs refer to noise propagation against and with the direction of flow respectively.

Noise generated by flow through a silencer is often referred to as 'regenerated noise'. It results from turbulence generated over the face of the acoustic lining and, where there are splitters in the airflow, by turbulence shed from the splitter. If the air velocity through the silencer passageways is appreciable then noise can be generated by the mixing of the high velocity jet flow just downstream of the silencer. The absolute sound levels are broadly a function of the airflow velocity through the silencer passageways raised to about the sixth power. Most silencer manufacturers publish data on the induct regenerated sound power levels. It is essential that this information is studied since if a silencer which is too small is used the effect can be to increase rather than decrease the induct sound levels.

5.2.5. Assessment of Performance—Insertion Loss and Transmission Loss

So far the concept of silencer performance has been used without formal definition. In terms of standardisation two different performances may be quoted, one being called the transmission loss and the other the insertion loss. The transmission loss is the difference (in dB) between the sound power level incident on the silencer and the sound power level transmitted out of the silencer and is an absolute, but effectively non-measurable, quantity. The insertion loss is the difference in sound power level at a given location in the ducting downstream of the silencer, due to the insertion of the silencer in place of an equal length of plain ducting. The actual values of insertion loss depend on elements of ducting connected to both ends of the silencer and can thus vary from test rig to test rig, from installation to installation. The British and International test Standards effectively measure the insertion loss in a standardised form of test rig such that the measured insertion loss should closely approximate the ideal transmission loss. However, the actual insertion loss of a silencer installed on site will be different from the insertion loss established in a standard test. The achieved silencing will be influenced by noise reflections from bends, junctions, controls, etc., in the ducting and also by the source of the noise itself. It is only quite recently that the influence of such effects has been appreciated and a great deal of work will be required before the effects are fully understood and quantified. As an example of the potential magnitude of such effects the following table shows how the insertion loss of one cylindrical silencer fitted with a centrebody varied as the blade pitch of an axial fan was altered.

MEASURED SILENCER INSERTION LOSS—dB

Blade pitch setting	Frequency — Hz						
	63	125	250	500	1000	2000	4000
Low	8	11	16	30	39	35	32
Medium	8	11	16	27	32	32	29
High	8	11	16	24	23	23	24

Fig. 5.8.

Dynamic Insertion Loss is measured using a sound power of high level to ensure that the regenerated noise does not influence the readings. Usually attenuator manufacturers quote dynamic insertion loss and regenerated noise levels separately. However, some manufacturers quote dynamic insertion loss data for various input fan sound power levels which thus takes into account regenerated noise.

5.2.6. Aerodynamic Performance—Pressure Drop Characteristic

The installation of a silencer in an airway generally increases the system pressure loss. Silencers for which there is no change in duct cross section area, i.e., cylindrical or rectangular silencers with no pods or splitters, impose an increased pressure loss only through an increase in wall friction as compared with a plain duct. Silencers with splitters and pods have generally much larger losses. There are pressure losses associated with the acceleration and deceleration of flow at the inlet and outlet of the splitters as well as increased friction losses arising both from an increased surface area and an increased velocity. The loss at the splitter outlet can be reduced by tapering the rear ends of the splitters so that there is a controlled diffusion. Similarly, rounding the inlet face of the splitters will reduce the inlet loss. (Pressure loss data for the silencing equipment supplied, and at the specified airflow rate, is available from the manufacturer.)

5.2.7. Installation and Location of Silencers

Silencers are used to control ductborne noise. Ductborne noise need not only come directly from the fan but may include noise generated by the flow of air across or through elements such as bends, control dampers or even silencers themselves, and noise entering the system through the walls of the ducting. Care must be taken therefore to ensure that noise is not 'short-circuited' across a silencer.

When silencers are close-coupled to a fan it is worth noting the points made in Fig 4.5. for a centrifugal fan. It is important that the airflow into a fan is as smooth and evenly distributed as possible. Particularly with axial fans, some caution should be exercised when close coupling a podded inlet silencer.

5.3. Control of Breakout Noise—Lagging

Most ducts are made of relatively lightweight metal and, if high sound pressure levels have been generated by the fan, appreciable sound energy can be radiated through the duct walls. In addition, highly turbulent flows, which may be generated downstream of a bend or obstruction, can induce a drumming noise from the duct wall. The following equation may be used to estimate the radiated sound power level Lw_{rad}' resulting from the breakout of noise from a length L of duct of perimeter P and cross section area S with a transmission loss* TL, provided the transmission loss is greater than about 10 dB.

$$Lw_{rad} = Lw_{in}\ 10 \log_{10} 1 - \exp \frac{-LP}{S \times (10^{TL/10})}$$

dB re 10^{-12}W.

Lw_{in} is the sound power level of the ductborne noise entering the section.

As an approximation the transmission loss of a simple wall is a function of its mass per unit area and frequency, with the relationship having become enshrined in what has become known as the 'mass law' or, more properly, the 'limp-wall mass law'. For a limp wall the random incidence loss increases by about 5 dB per doubling of surface mass or per doubling of frequency.

For ducting of rectangular cross section the 'mass law' transmission loss can be used in the above equation with reasonable accuracy.

For straight lengths of circular ducting the low frequency transmission loss can be considerably greater than the mass law would predict, even for lightweight flexible ducting.

Lagging ductwork will reduce the level of breakout noise, but it should be noted that all the exposed ducting must be treated. The fitting of an attenuator is probably a more cost-effective approach. Typically, lagging would consist of a layer of glassfibre or equivalent porous material of the order of 50-100 mm thick covered by a limp impermeable membrane, such as neoprene or leaded vinyl sheet, with a surface mass of around 2.5 kg/m^2. This treatment would give an additional wall transmission loss of around 12 dB at 250 Hz and 36 dB at 1000 Hz. It may be noted that this construction provides an improvement in transmission loss greater than that which would be predicted by the mass law for the addition of the extra mass. Glassfibre matting or a similar type of thermal lagging used on its own without a heavy wrapping will not give any appreciable increase in transmission loss, while the use of heavier ducting or the application of proprietary damping compound will produce improvements only in line with the mass law — requiring substantial increases in mass to achieve significant noise reduction.

5.4. Enclosures

An acoustic enclosure is essentially a box placed around a noisy machine. The noise radiated from the machine is attenuated in its transmission through the walls of the enclosure. Enclosures can vary in construction from being built of brick, through a whole range of proprietary panels to simple wooden frameworks with plywood panels on one side and limp sheets on the other with acoustic infill between. Enclosing the unit will result in high sound pressure levels being built up within the enclosure and some sound absorbent lining treatment may be necessary if access were required while the machine was operating. If the enclosure does not fully surround the noise source, its acoustic performance is very severely downgraded. By way of example, an opening of one per cent of total area in an enclosure with walls giving a 30 dB transmission loss would reduce the effective transmission loss to 20 dB.

An important consideration when fitting an enclosure is the provision of adequate ventilation for cooling air to the electric motor and drive. Often this may require the addition of cooling fans which, in turn, require to be silenced.

*Footnote — In this context, the transmission loss is the difference in dB between the sound intensity incident on to a section of duct wall and the sound intensity radiated away from that section, i.e., it is the loss of energy across the duct walls.

6. FAN VIBRATION

6.1. Importance of Balance

In common with all machinery, rotating or reciprocating, a fan will vibrate to some degree when it is in operation. The vibration may be due to one or more causes, particularly when it is installed in a system. It may arise from the fan itself; excessive turbulence in the airstream, especially ahead of the fan; unsuitable or incorrectly installed anti-vibration mountings; structureborne vibrations or the fan operating against excessive resistance to airflow. If a fan is judged to have an unacceptable vibration level it must first be established that the fan is correctly installed. This could mean that the fan is bolted to a solid floor or is installed on spring mounts with flexible connections to its associated duct work.

Having ascertained that the source of excessive vibration is the fan, the cause must be identified. Drive systems, belts and bearings, and all fixings must first be checked before it can be assumed that the fan is out of balance. Even then it has to be established that all rotating assemblies are operating well below their critical speeds.

Out-of-balance is the commonest and most important source of vibration in fans. The resulting vibration occurs at the rotational frequency of the fan. It may be acceptable or excessive as judged by the relevant standard BS 4675. Other sources of vibration, such as air turbulence, do arise in fans but are not generally significant and are not considered in this document.

Excessive vibration will have a detrimental effect on the trouble free operation of the fan. Bearing life will be shortened particularly if the fan is on a solid mount. Fixings may gradually loosen and eventually result in a catastrophic failure and unacceptable fluctuating loads will be transmitted to the supporting structure.

It is thus important to ensure that the fan is balanced correctly for the class of machine, its speed of rotation and application.

6.2. Static and Dynamic Unbalance

Two forms of balance defect can be detected: static unbalance and dynamic or couple unbalance.

Static unbalance simply means that the rotor has a heavy side which can be exactly counterbalanced by a suitable mass placed on the opposite side at the same axial position along the rotor.

Though a rotor may be in perfect static balance it can still produce a vibration at its bearings due to a nodding motion. For instance, in an elongated rotor a different static balance condition, at either end, will result in a significant residual couple unbalance, this couple unbalance being the cause of the nodding motion.

Fig 6.1. is a diagrammatic illustration of the two types of unbalance indicating where counter-balance weights may be needed. If trim balancing is required on site, a successful result can often be achieved by placing balance weights in one plane

TWO TYPES OF ROTOR UNBALANCE

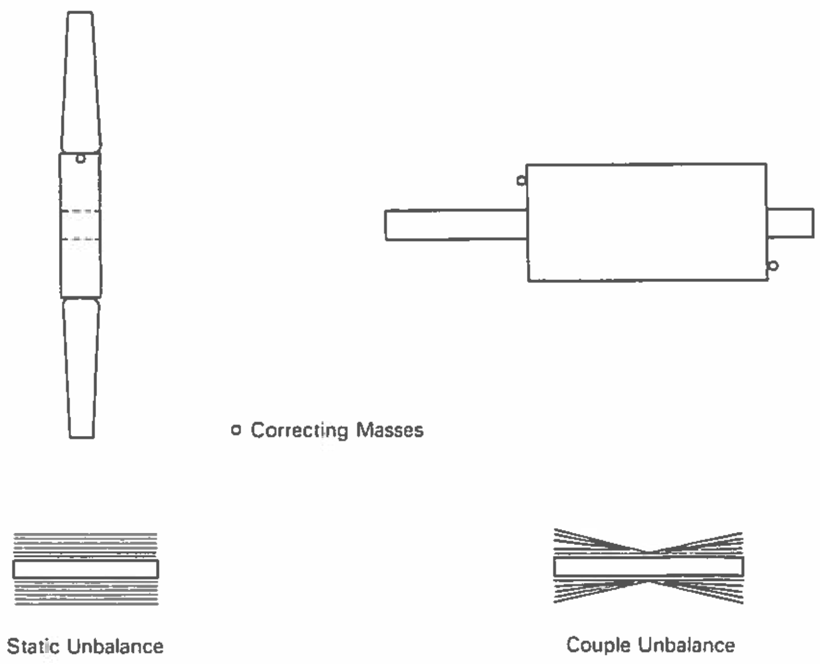


Fig. 6.1.

only, in other words, static balancing. Usually the balance weights are added as extra washers under a nut or screw in the balancing plane.

Portable balancing equipment is available from a number of manufacturers, making on-site balancing a straightforward operation. Nonetheless, it is advisable to call in a specialist balancer, or the manufacturer, if as a result of verifying the points outlined in Section 6.1. it has been established that fan balance is causing the unwanted vibration.

If, for any reason, it becomes necessary to change the whole, or part, of the fan's rotating assembly, or its bearings, it is advisable to re-check the fan vibration level before putting the fan back into service.

6.3. Principles of Measurement

Fan unbalance manifests itself as a periodic vibration characterised by a sine wave — the so-called simple harmonic motion. With suitable instrumentation, there are three properties of the fan vibration which can be directly measured: displacement, velocity and acceleration. The measurement of each of these three properties has its purpose.

Displacement: this is often measured to quantify a subjective assessment of vibration level. It is important when considering the maintenance of running clearances.

Velocity: agreed as the best general measure of vibration level and used by ISO for classifying vibration severity levels for different types of machine. For a given class of machines a single vibration velocity figure can be specified over a wide range of speeds, resulting in satisfactory service.

Acceleration: the measure which gives rise to forces and stresses within a machine, and between a machine and its foundation.

6.4. Relationship between Vibration Properties

The symbols below are used to explain the relationship between displacement, velocity and acceleration, for a periodic vibration characterised by a sine wave:

- x Instantaneous displacement
- X Maximum displacement from structure equilibrium peak
- t Time
- v Velocity
- a Acceleration
- w Angular frequency ($2\pi f$)
- f Vibration frequency

Displacement

$x = X_{peak} \sin(wt)$

Velocity

$v = w X_{peak} \sin(wt + \pi/2)$

Acceleration

$a = w^2 X_{peak} \sin(wt + \pi)$

These equations show that the properties all behave as a sine wave and that the velocity is advanced in time by one quarter cycle and the acceleration by one half cycle in relation to the displacement.

6.5. Units of Measurement

Displacement: measured in micro-metres with the symbol μm (i.e. m x 10^{-6}). The peak value is normally quoted.

Velocity: measured in millimetres/sec or mm/s; both the PEAK and RMS (root mean square) values are used. However, the RMS value is a better descriptive quantity owing to its relationship to the energy content of vibrations.

Acceleration: measured in metres per sec per sec or m/s^2, but often normalised by expressing the results in relation to a different reference level. Often acceleration is referred to in units of standard gravity, $9.81 m/s^2$. Sometimes the decibel scale, referred to 10^{-5} m/s^2, is used:

$20 \log_{10} \frac{a}{10^{-5}}$, where 'a' is in m/s^2. Fig 6. 3. shows acceleration plotted against velocity and speed of rotation.

6.6. Conditions of Measurement

The machine support may have a significant effect on the vibration levels recorded on a machine under test. It is important therefore that when a vibration level is specified the machine mounting is also described.

To take an obvious example, if a machine is 'solid' mounted, the vibration level recorded will be considerably less than if the same machine were mounted on high deflection spring mounts.

When a vibration specification is related to a rigidly mounted machine it should be appreciated by the user that all the out-of-balance forces are entirely reacted at the bearings. For comparable service life, a rigidly mounted machine would require a lower vibration level than the same machine soft mounted.

Generally speaking, the operating conditions of the fan under test, for example temperature, speed and load, are not specified. Under these circumstances the fan is usually tested with no air resistance and at full speed only. However, there are many special purpose fans where the operating conditions are laid down in the vibration specification. This can be very important where variable speed machines are involved.

6.7. Standards

There are two British Standards, both having an ISO equivalent, which relate to fans. One is concerned with the balance quality of rotating components and the second, vibration standards of complete machines.

6.8. Component Balancing

The relevant British Standard is: BS 5265 : Part 1 : 1979 : Mechanical Balancing of Rotating Bodies: Part 1. Recommendations on Balance Quality of Rotating Rigid Bodies. ISO 1940-1973.

VIBRATION PARAMETERS

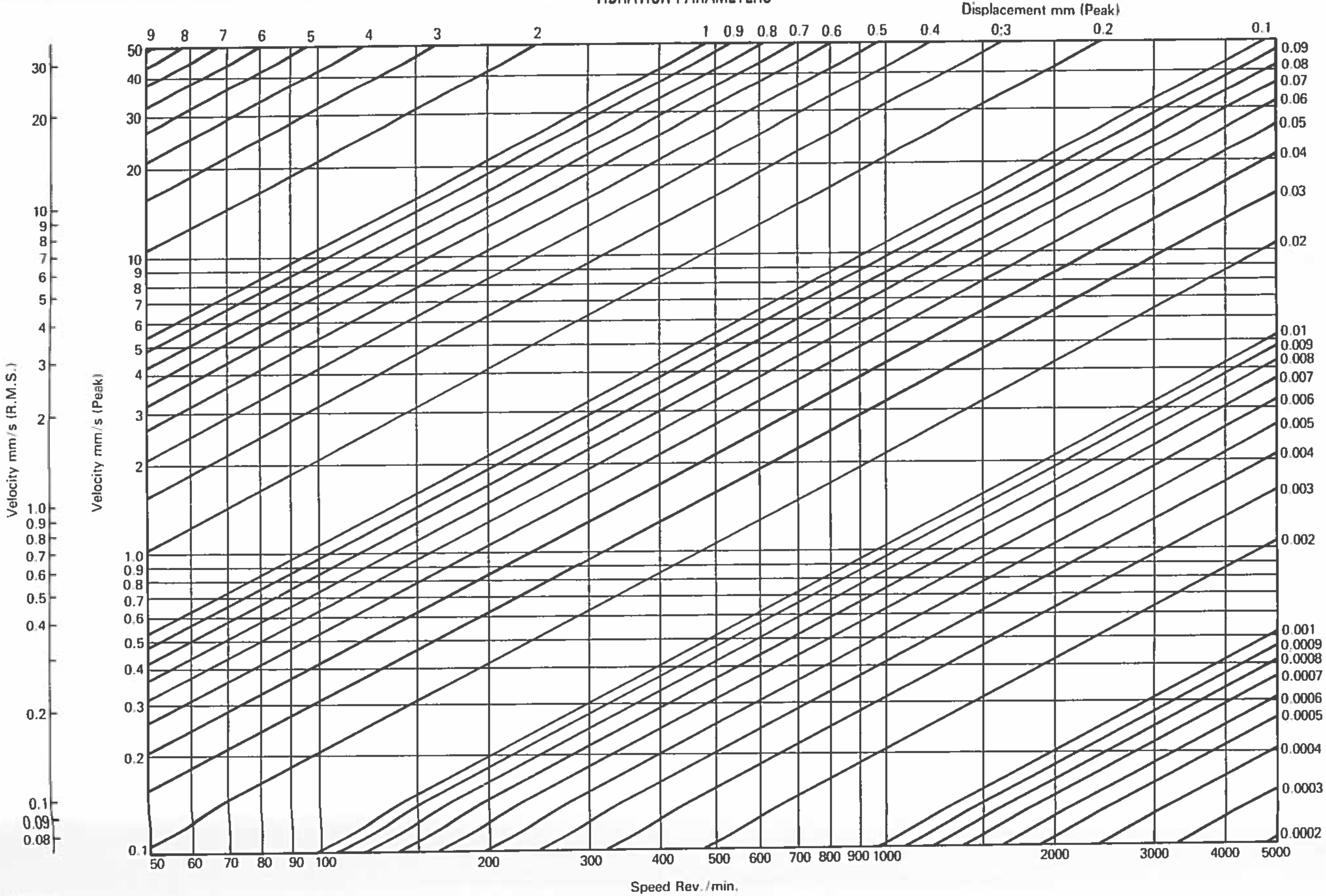


Fig. 6.2.

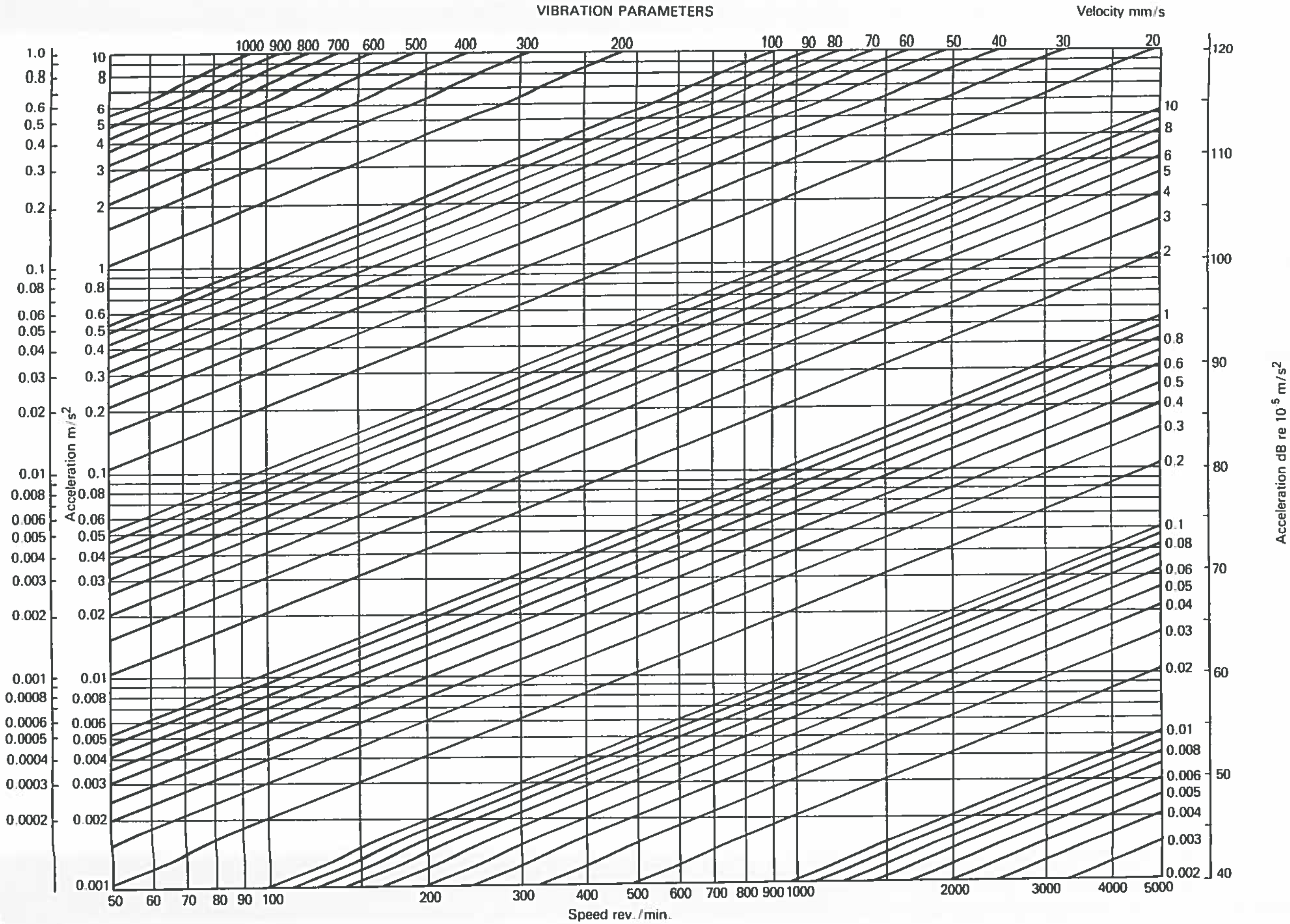

VIBRATION PARAMETERS
Velocity mm/s
1000 900 800 700 600 500 400 300 200 100 90 80 70 60 50 40 30 20
10 8 6 5 4 3 2 1 0.8 0.6 0.5 0.4 0.3 0.2 0.1 0.08 0.06 0.05 0.04 0.03 0.02 0.01 0.008 0.006 0.005 0.004 0.003 0.002
Acceleration g
1.0 0.8 0.6 0.5 0.4 0.3 0.2 0.1 0.08 0.06 0.05 0.04 0.03 0.02 0.01 0.008 0.006 0.005 0.004 0.003 0.002 0.001 0.0008 0.0006 0.0005 0.0004 0.0003 0.0002
Acceleration m/s²
10 8 6 5 4 3 2 1 0.8 0.6 0.5 0.4 0.3 0.2 0.1 0.08 0.06 0.05 0.04 0.03 0.02 0.01 0.008 0.006 0.005 0.004 0.003 0.002 0.001
Acceleration dB re 10⁻⁵ m/s²
120 110 100 90 80 70 60 50 40
Speed rev./min.
50 60 70 80 90 100 200 300 400 500 600 700 800 900 1000 2000 3000 4000 5000

Fig. 6.3.

BS 5265 sets out a range of balance quality grades for rotors for different types of machine, each main grade being separated from the next by a factor of 2.5. The grade recommended for fan rotors is G 6.3 or a PEAK vibration velocity of 6.3mm/s.

For uses which are less or more sensitive to balance than the general purpose application, vibration levels which are above or below the 6.3mm/s PEAK vibration velocity may be specified.

Fig 6.2. shows the relationship between the balance quality grades relevant to fan rotors, the residual unbalance in terms of centre of gravity displacement and speed of rotation.

It should be noted that an assembled, but otherwise rigid, rotor made up from two or more components, for example a fan impeller and the rotor of an electric motor, may have a balance quality substantially different from G 6.3 even if all components have been separately balanced to G 6.3. This can arise in two principal ways:

(i) misalignment and lack of concentricity between rotor components.

(ii) different degrees and phases of residual unbalance in the rotor components.

6.9. Machine Vibration

The relevant British Standard is: BS 4675 : Part 1 : 1976 : Mechanical Vibration in Rotating and Reciprocating Machinery: Part 1. Basis for specifying Evaluation Standards for Rotating Machines with Operating Speeds from 10 to 200 Revolutions per Second. (ISO 2372)

BS 4675 is concerned with complete machines and proposes rules for the evaluation of the vibration of 'normal' machines in relation to safety, reliability and human perception.

Table 6.1. is an extract from Annex A of BS 4675 and shows examples of quality judgement for different classes of machines and vibration severity ranges. It should be noted that 'vibration severity' is defined as the maximum RMS value of vibration velocity measured at significant points of a machine, such as a bearing or mounting point.

For general purpose fans and in the absence of any specific requirement on vibration level, it is recommended that a vibration severity of 4.5mm/s is not exceeded on a new fan. For special purpose fans or in sensitive application a vibration severity no greater than 2.8mm/s is suggested.

In service, the vibration level of a fan will usually deteriorate as bearings wear and uneven deposits build up on rotating parts. This process should not be allowed to go too far, however, as a catastrophic and expensive failure could ensue. If the vibration level reaches more than 60% above that when the fan was commissioned, the fan should be shut down as soon as possible and the cause determined.

6.10. Positions and Modes of Vibration Measurement

Readings of vibration on a machine which is supported on anti-vibration mountings should be taken immediately adjacent to the mounts, on the fan side. If the machine is solid mounted, readings should be taken as close as possible to the bearings. In order to check for couple unbalance a vibration reading must be taken at both ends of the fan, whether it is soft or solid mounted. In no case should the reading exceed the specified level.

For most fans it is sufficient to achieve the specified vibration severity when measurements are taken at right angles to the shaft axis and in the direction of least mounting stiffness. In other words, in the vertical direction on a horizontally, soft mounted machine. On a solid mounted machine it is advisable to check the vibration in the vertical and horizontal direction.

As there are rarely any significant unsteady forces in the fore and aft direction, it is only in exceptional circumstances that the vibration parallel to the shaft axis need be monitored. For the vast majority of fans this mode of vibration can be safely ignored.

Table 6.1.

Vibration Severity Ranges and Examples of their Application to Small (Class I) Medium (Class II) and Large Machines (Class III)

Ranges of Vibration Severity: Vmm/s RMS		Examples of Quality Judgement for Separate Classes of Machine		
Range	Range Limits	Class I	Class II	Class III
0.45				
	0.45			
0.71				
	0.71	A		
1.12				
	1.12		A	
1.8				
	1.8			A
2.8				
	2.8		B	
4.5				
	4.5	C		B
7.1				
	7.1	D	C	
11.2				
	11.2		D	C
				D

At present experience suggests that the following classes are appropriate for most applications.

Class I: Individual parts of engines and machines, integrally connected with the complete machine in its normal operating condition (Production electrical motors of up to 15kW are typical examples of machines in this category).

Class II: Medium-sized machines (typically electrical motors with 15 to 75kW output) without special foundations, rigidly mounted engines or machines (up to 300kW) on special foundations.

Class III: Large prime movers and other large machines with rotating masses mounted on foundations which are relatively soft in the direction of vibration measurement (for example, turbo-generator sets, especially those with light-weight substructures).

7. VIBRATION ISOLATION

7.1. Why Vibration Isolation

In its broadest sense vibration isolation infers the provision of a break in the flow of vibrational energy from one point to another. The practical meaning of this statement is that the sources of vibration, usually an out-of-balance rotating assembly, are isolated from the structure to which the machine containing the out-of-balance rotor is attached.

Apart from the basic function outlined above, vibration isolation serves to protect the structure or building from the effects of vibrating equipment and, at the same time, reduces re-radiation of vibrational energy causing unnecessarily high room sound pressure levels.

7.2. Fundamentals

The vibration break referred to above usually takes the form of a spring, rubber or neoprene. The material of the spring, however, is chosen for its particular characteristics and to suit the application.

Considering the case of a fan mounted on springs, if one pushes the fan up and down at low frequency, movement of the fan and therefore deflection of the isolator is resisted by the stiffness of the isolator/spring. Thus, if the spring stiffness, usually given in kg/mm, is k and the deflection of the mount is δ mm then the force F_s required to deflect the isolator is:

$$F_s = \delta k$$

The above relationship is dominant at low frequencies. At higher frequencies, inertia of the fan starts to influence the resistance to movement and at very high frequencies inertia is predominant. In this case the relationship between inertia and amplitude of movement is:

$$F_i = \frac{-\mathrm{Iw}\,(2\pi\, f)^2\delta}{g}$$

where Iw is the weight of the fan and f is the frequency of vibration.

The inertia force acts in the opposite direction to the spring force and this is the reason for the negative sign.

It can be seen that the inertia force increases rapidly with frequency whereas the spring force is independent of frequency. Obviously, at some frequency, the two restraining forces will cancel out. This frequency is called the natural frequency and, in theory, zero force is required to produce very large movements. In practice the movements would be restricted by some form of damping.

Combining the two equations above results in the following expression for the natural frequency f_n in Hz:

$$f_n = 15.8\sqrt{\frac{1}{\delta o}}\ \mathrm{Hz} \qquad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

It should be noted that the natural frequency depends solely upon the static deflection of the spring resulting from the weight of the fan. In the above relationship δ_o is measured in mm and the equation is shown graphically on Fig 7.1.

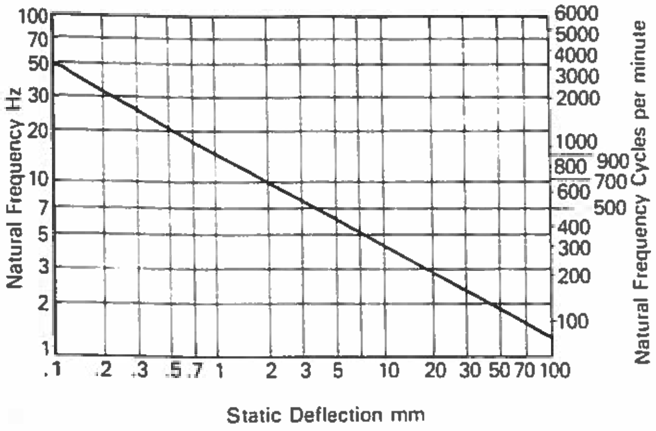


Fig. 7.1. Effect of static deflection on natural frequency

The significance of the natural frequency is that if a fan is running with an out-of-balance impeller at a speed which coincides with the natural frequency then very large movements would occur. The movement would only be limited by damping or something breaking, the condition being known as resonance.

7.3. Transmission & Isolation

The effectiveness of a vibration isolation system is usually expressed in terms of isolation efficiency or transmissibility. Often, these two terms are expressed as a percentage.

The degree or efficiency of the isolation is related to the proportion of the disturbing force which is transmitted, in other words the Transmissibility.

$$\text{Transmissibility } T = \frac{F_t}{F_o}$$

where F_t is the transmitted force and F_o the disturbing force.
Correspondingly, Isolation Efficiency is:

$$\frac{F_o - F_t}{F_o} = 1\text{-}T$$

Transmissibility is the more practical term to use as it indicates directly the proportion of the disturbing forced fed into the supporting structure.

Table 7.1. below serves to illustrate the point.

(1-T)%	T%	F_o (kg)	F_t (kg)
0	100	100	100
70	30	100	30
80	20	100	20
90	10	100	10
99	1	100	1

An Isolation Efficiency (1-T) of 90% is good and an efficiency of 99% may not seem worthwhile going for until it is realised that the transmitted force F_t has reduced by a factor of no less than 10. This can be seen in the last two lines of the table.

7.4. Damping

A vibration isolation system is influenced by three factors: the stiffness of the isolation system, the inertia of the equipment being supported and the damping inherent in the isolators. The first two factors have already been discussed. The third, damping, has two beneficial effects but one on the debit side:

(a) the effect of resonance is reduced;
(b) movement of the suspended system is reduced at all frequencies, and
(c) the transmitted force is increased.

The diagram below shows the effect of a damped and an undamped system on a free vibration. In (a) the vibration of the undamped system will, under ideal conditions, continue unabated as no energy is being taken out of the vibrating system. In (b) damping is present, energy is being progressively removed from the system and the vibration amplitude is being steadily reduced.

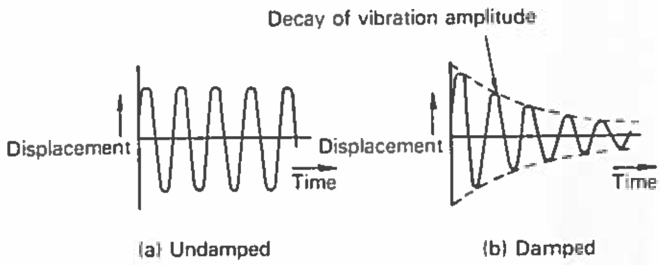


Fig. 7.2. Free Vibration and Damping

If the amount of damping present is just sufficient for the isolated mass to return to its mean position in minimum time then the system is said to possess "dead beat" or "critical" damping. The ratio of actual damping present to that for critical damping is referred to as the damping ratio or D.

For systems where little or no damping is present, transmissibility becomes:

$$T = \frac{1}{\left(\frac{f}{f_n}\right)^2 - 1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

where f is the frequency of the disturbing force.

Where damping 'D' is present:

$$T = \left\{\frac{1 + 4D^2\left(\frac{f}{f_n}\right)^2}{\left(1 - \left(\frac{f}{f_n}\right)^2\right)^2 + 4D^2\left(\frac{f}{f_n}\right)^2}\right\}^{1/2} \quad \ldots\ldots\ldots(3)$$

In practice, damping ratios for most fan isolating systems are very small and can, for practical purposes, be ignored. Typical values are:

Isolating Material		D
Natural Rubber	70 Shore	.04
Natural Rubber	40 Shore	.02
Steel Springs		.005

The effect of various degrees of damping is illustrated in Fig 7.3.

As well as showing the effects of damping Fig 7.3. indicates that no isolation can occur until the disturbing frequency is greater than 2 x the natural frequency.

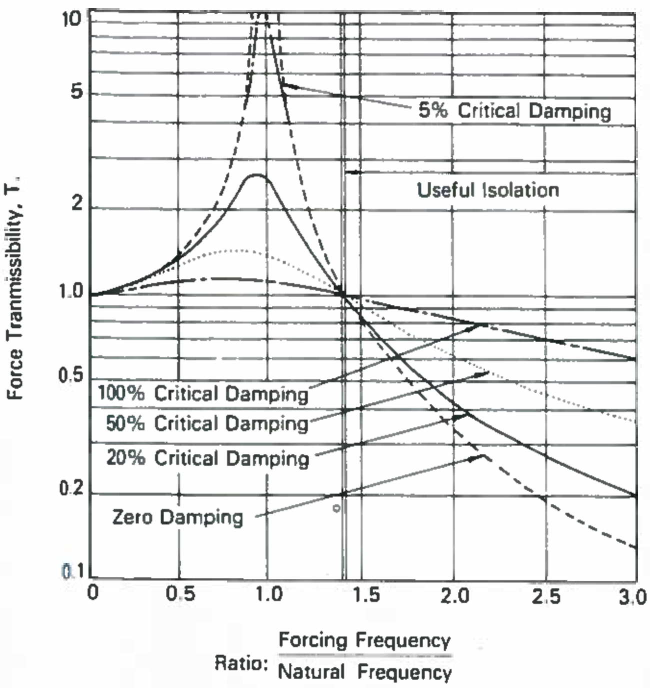


Fig. 7.3. Effect of Damping on Force Transmissibility

A closer look at Fig 7.3. will show that practical Transmissibilities, 30% or less, and therefore Isolation Efficiencies of 70% or more, are not achieved until the frequency ratio is greater than 2.

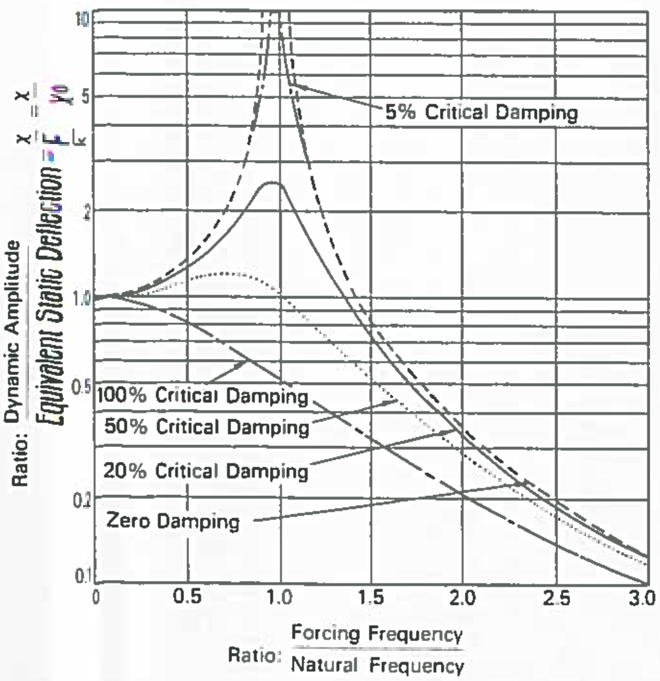


Fig. 7.4. Effect of Damping on Amplitude of Motion

As indicated earlier, damping also reduces the amplitude of motion of the suspended system. This is shown in Fig 7.4. where the relation:

$$\frac{\delta}{\delta_o} = \left\{ \frac{1}{\left(1 - \left(\frac{f}{f_n}\right)^2\right)^2 + 4D^2 \left(\frac{f}{f_n}\right)^2} \right\}^{1/2}$$

is shown against f/f_n for various values of the damping ratio, δ is the dynamic amplitude, whilst δ_o is, as before, the static deflection.

In the region of interest, namely $f/f_n > 2$, it will be seen that increased damping reduces the operating amplitude. In contrast, the reverse effect on transmissibility occurs where increased damping has a detrimental effect.

The figure also shows that practical levels of damping have a negligible effect on the operating amplitude at the commonly used frequency ratios.

7.5. Sources of Vibration

Fans fall into the general classification of rotating machinery. As discussed in Chapter 6, the most significant source of vibration, and usually the one requiring to be isolated from the structure, is out-of-balance of the rotor. The frequency of this vibration will usually be the fan RPM/60 Hz. Higher 'harmonics' also appear. In other words, a significant vibration level can be measured at 2 or 3 times the fan rotational frequency.

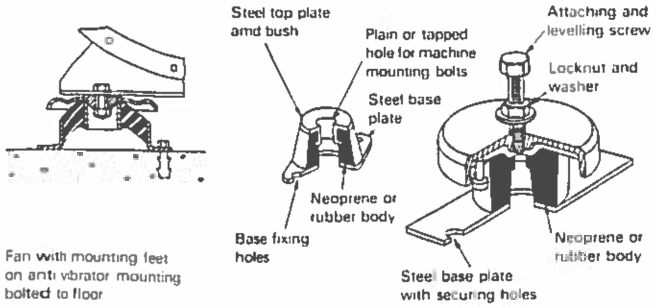
Though a fan is carefully balanced by the manufacturer, the quality of balance, and therefore degree of vibration, can deteriorate in service due to a number of causes: slackness in bearings can develop due to wear, blades can wear unevenly from erosion and deposits can build unsymmetrically on the blades because of local 'throwing off' of deposited material.

In addition to the foregoing aspects, a bad 'aerodynamic' installation can cause unwanted vibration. Whilst this aspect is dealt with in detail in the HEVAC Fan Application Guide, it is important to note here that a disturbed inlet flow or an excessive system pressure requirement will result in undesirable vibration amplitudes in the direction of the shaft axis. This is particularly so in axial flow machines. If this form of vibration is evident, then the installation must be examined and the cause eliminated. There should generally be no need to provide vibration isolation in the direction of the shaft axis.

Where a fan is mounted with its shaft axis vertical, then the vibration to be isolated will be at right angles to the mount load bearing direction and the isolator selection must take this into account.

Higher frequency vibrations or 'noise' can be of magnetic origin and will be satisfactorily isolated from the fan supporting structure provided care has been taken of the fan vibrations at rotational speed.

Magnetic noise radiated by the fan structure, whilst always of motor origin, can be the result of incorrect motor assembly or the waveform characteristic of the electrical supply. In either case the problem should be referred to the fan supplier.



Rubber or neoprene in shear

Fig. 7.5.

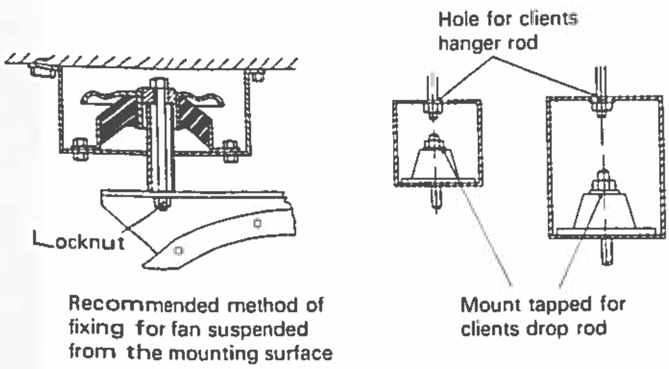
7.6. Isolator Types & Construction

A number of different types of isolator are used with fan equipment and, though application areas overlap, each type has its own general purpose.

The simplest type of isolator is the so-called mat or pad. Usually made from cork or synthetic rubber, a typical deflection would be about 5mm. Though used as part of an isolating system, these materials are not generally used on their own to isolate fan equipment from the supporting structure.

The most commonly used isolator is the rubber-in-shear/compression type. Attachment is both to the fan and the supporting structure and a typical maximum deflection is 15mm.

To form the mount Neoprene or rubber is bonded to suitably shaped metal surfaces in which provision is made for the fixings. Popular designs are shown in Fig. 7.5. The application of these designs would be for fan equipment supported from below. In many cases, particularly with axial flow fans, the fan is suspended/hung from above. For this type of installation special arrangements, often incorporating a 'standard' type of mount, are available. Typical examples are shown in Fig. 7.6. In both cases, a standard mounting foot and a standard anti-vibration mount can be used.

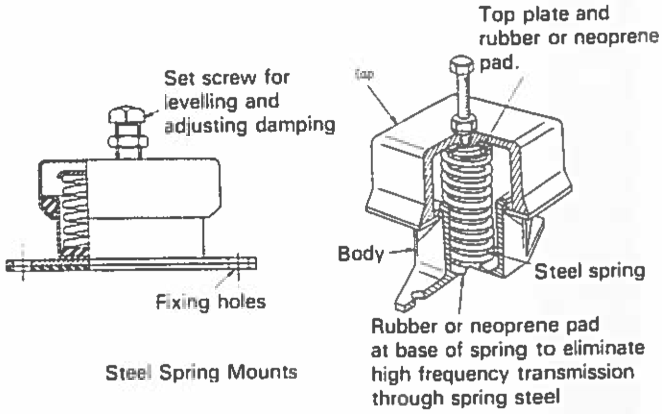


Rubber or neoprene in shear suspended mounting

Fig. 7.6.

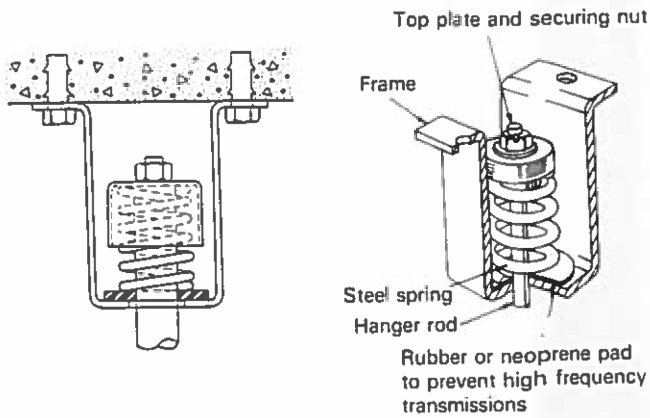
Where larger deflections, or small Transmissibilities at low frequencies are required a steel spring mount is necessary. These are available for deflections up to about 175mm. Because of the high deflection capability, these mounts are usually provided with a levelling bolt, to allow for variations in the calculated loading for each mount. Another feature normally provided with a spring mount is a ribbed Neoprene or rubber pad of low deflection. This is to prevent vibration/noise at acoustic frequencies, which travels down the spring, passing into the support structure.

Steel spring mounts incorporating the above features are shown in Fig 7.7. As with the rubber-in-shear mounts, they can be used in suspended arrangements, examples being shown in Fig 7.8.



Steel Spring Mounts

Fig. 7.7



Steel Spring Mounts
Suspended Mountings

Fig. 7.8.

7.7. Isolator Selection

The preceding sections have dealt with the reasons for the need to provide vibration isolation, the theory of isolation, isolator types and their principal characteristics. It now remains to select and install the optimum isolator, or isolating system, for the equipment in its operating environment.

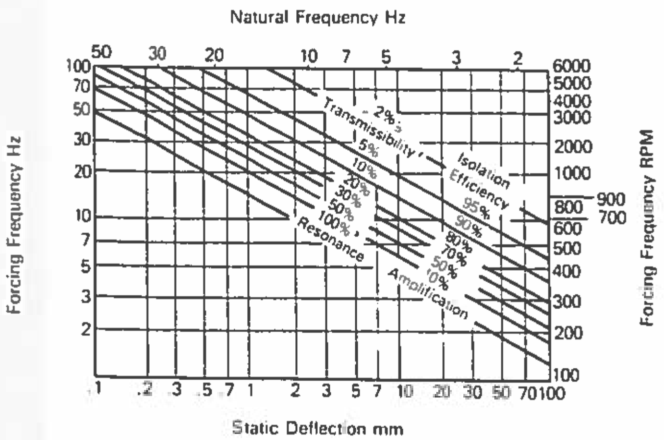
Before selecting the isolator an acceptable transmissibility has to be determined. Due account must be taken of the size of fan, its operating speed, supporting structure and location: for example, is it installed in the basement of a building or in a roof-top plant room.

Guide lines have been determined for recommended transmissibilities for various types of fans and these are shown in Table 7.2.

Table 7.2.

Sensitive Locations	% Trans-missibility	Less Sensitive Locations	% Trans-missibility
Centrif. Fans >20kW	2	"	10
Axial Fans >40kW	4	"	20
Centrif. Fans 4 to 20kW	4	"	20
Axial Fans 7.5. to 40kW	6	"	25
Centrif. Fans up to 4kW	6	"	30
Axial Fans up to 7.5kW	10	"	30

Using the appropriate relationships already established, the connections between transmissibility, forcing frequency, natural frequency and static deflection can be illustrated graphically as shown in Fig 7.9. For example: a 15kW centrifugal fan, running at 1170 RPM and located at basement level should have a transmissibility of 20%. The static deflection of the desired isolators, from the Fig 7.9., should therefore be a minimum of 3mm. If the same fan were mounted in a critical location the desired deflection would be 14mm — considerably greater.



Performance of lightly damped mounts

Fig. 7.9.

In those instances, and they are more and more common, where the fan is mounted other than in the basement or on the ground floor of a building or other structure, the support for the fan can be far from rigid. Nowadays, structural engineers recommend that the maximum deflection due to the load at the centre of a span must not exceed 1/250 of the distance between supports. On the other hand, a safety factor in the region of 5 is also incorporated into many structural designs. Combining these two factors together with the equation relating static deflection and natural frequency leads to a further table: — 7.3.

Table 7.3.

Span	Allowable Deflection	Likely Deflection	Minimum Likely f_n	Minimum Likely f_n
m	mm	mm	Hz	cpm
3	12	2.4	10.0	600
5	20	4.0	8.0	480
10	40	8.0	5.5	330
12	48	9.6	5.0	300
15	60	12.0	4.5	270

As a direct result of supporting structures becoming more flexible, the situation has arisen where it is possible for the fan, particularly if it is large and slow running, to be in resonance with the floor system. In this circumstance a very small out-of-balance force will result in an unacceptably large floor movement.

The above natural frequencies are quoted as minima as they refer to the span being loaded at the centre. If the equipment is not located centrally then the natural frequency would be higher. However, the point must be appreciated that the structure on which the fan is mounted is often not strictly rigid. It has a design deflection and associated natural frequency with the specified loading.

To illustrate how easy it is to select the wrong mount for the application, consider a 15kW fan running at 1200 RPM and mounted centrally on a floor of 5m span. The fan is located in a non-sensitive area and from Table 7.2. a transmissibility of 20% is appropriate. If reference is now made to Fig. 7.3. it will be readily seen that for an isolator with 5% damping a frequency ratio of 2.5 is required to give the desired transmissibility. In other words, the isolator natural frequency should be:

$$\frac{1200}{60 \times 2.5} = 8 \text{ Hz}$$

But this is just the same as the natural frequency of the supporting floor. Although the dynamics of the total system would affect the final result, some 20% of the rotational out-of-balance force could be fed into the fan support structure, at its resonant frequency, causing a movement many times that originally intended. To avoid this kind of situation occurring, where resilient structures are used, the deflection employed is usually such that the natural frequency of the isolator is less than half that of the supporting structure. In other words, the isolator has a deflection of 4 to 5 times that of the loaded floor.

Where fans are located in buildings, the trend is therefore developing such that suitable isolator deflections are recommended rather than specific transmissibilities. Some guidelines are given in Table 7.4.

Table 7.4.

Guideline Deflections to Avoid Resonances

Equipment (Mid Span Location)	Minimum Static Deflection mm Base ment	Above Ground Floor Span 6m	9m	12m	15m
Axial Fans (Floor Mounted)					
Up to 4kW	6	25	25	25	25
5 to 15kW, up to 500 RPM	12	40	50	50	60
above 500 RPM	12	25	25	40	90
Over 17.5kW, up to 500 RPM	20	50	60	70	90
above 500 RPM	12	25	30	40	50
Centrifugal Fans (Floor Mounted)					
Low Pressure (up to .75 kPa)					
Up to 4kW	6	25	25	25	25
Over to 5.5kW, up to 500 RPM	12	40	50	50	60
above 500 RPM	12	25	25	40	50
High Pressure (over .75 kPa)					
Up to 15kW, 175-300 RPM	9	60	60	90	120
301-500 RPM	12	50	50	60	90
above 500 RPM	9	30	30	50	60
Over 17.5kW, 175-300 RPM	40	60	90	120	140
301-500 RPM	25	50	60	90	120
above 500 RPM	12	30	50	60	90

Referring to the last example, Table 7.4. would suggest a static deflection of 30mm. Using Fig 7.1. it will be seen that the natural frequency of the isolator is 2.9Hz or 0.36 that of the floor which was 8Hz. The 30mm deflection would thus avoid any unexpected resonances.

To summarize:

1. If the fan is to be mounted in a building where the support structure is not strictly rigid then the relevant deflection should be selected from Table 7.4. and an isolator with the appropriate load capacity chosen.

2. Whilst observing the above will generally avoid vibration problems there are a great many cases where the cost of providing the deflections shown in Table 7.4. is unnecessary. There are numerous fans, particularly direct drive both centrifugal and axial, where simple low deflection 'rubber' mounts, possessing a significant degree of damping, are quite satisfactory. Where the rotor weight is small and the standard of balance good the out-of-balance forces are low and the isolators serve to remove bearing, air turbulence and magnetic frequencies. Selection of simple 'rubber' mounts is usually based on load capacity only, and no account is taken of the fan operating speed. The result is that many fans, in respect of isolation of the rotational speed, are operating in the region where some amplification of movement could take place. Nonetheless, and for the reasons stated, this rarely causes a problem and the higher frequencies are successfully isolated.

7.8. Installation

7.8.1. Having selected what appears to be the correct mount for the application, a number of other important aspects must be considered, and possibly acted upon, before a successful installation can be expected.

(i) Are the mounting points evenly loaded.

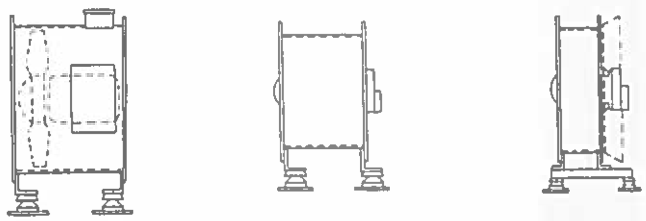
(ii) Is the fan stable on its mountings under service conditions and will it remain on its mountings under the influence of aerodynamic and torque reaction forces on start-up.

(iii) Is the mount selection appropriate if the fan shaft is vertical.

(iv) Are the isolators being 'bridged' by duct or service connections.

7.8.2. For catalogue selection purposes and installation the simplest approach is to have four equally loaded mounts. The fan installation will then be level and no muddles can occur on site due to the wrong mounts being put in the wrong place.

This approach can be readily achieved, within practical limits, on many direct drive fans. On some fans, for example, indirect drive units where the motor is mounted to the side of the fan, the support structure can be designed to even out the mount loads. Some sketches to illustrate these points are shown below.



Long Casing Medium Casing Short Casing

Axial Fan

Fig. 7.10.

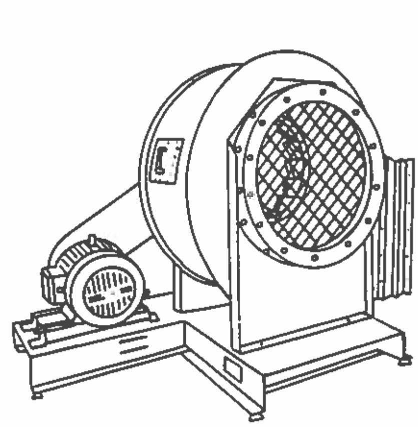
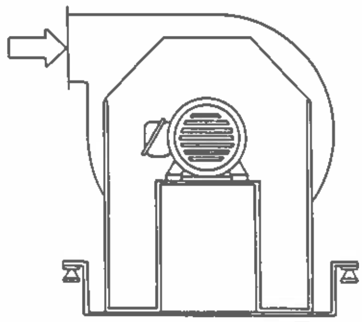


Fig. 7.11. Centrifugal Fan

It is important that the static deflection of each mount is almost the same. If this is not the case, a rocking motion of the machine could be set up with a frequency which is higher than that of the simple vertical mode.

7.8.3. From the vibration point of view, the vast majority of fans can be satisfactorily installed using simple mounting feet or a suitable base frame. Medium and low pressure direct drive fans and some belt drive axials fall into this category. However, a base frame is often needed for the medium pressure indirect drive centrifugals and the more powerful axials.

As shown in Fig 7.12 and 7.13 a steel base frame can be used to spread the mounting points to give even loading. It can also be used to lower the centre of gravity of the equipment. This latter feature is useful in terms of increasing the stability of high pressure centrifugal fans where the 'top horizontal' discharge configuration is used. Fig 7.12 illustrates the points.



Base Frame Design to improve stability

Fig. 7.12.

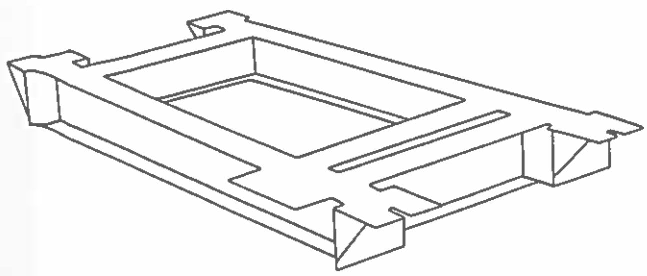


Fig. 7.13. Base Frame

In certain circumstances it is necessary to mount the fan equipment on a 'concrete inertia base'. Whilst there are good reasons for resorting to this type of installation it does have the disadvantage that the fan equipment cannot be completely factory assembled and tested as the concrete for the inertia base is normally poured on site. A typical inertia base is shown in Fig 7.14.

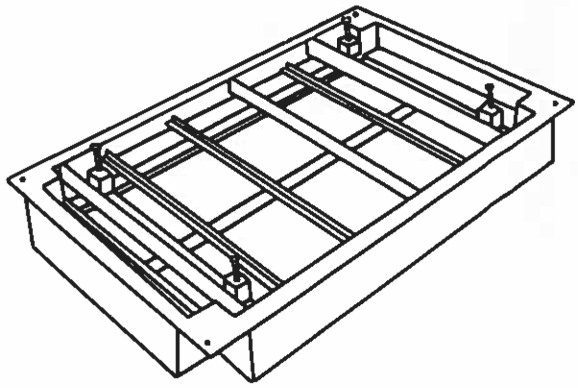


Fig. 7.14. Inertia Base

On occasions it will be necessary to have a very low level of transmissibility. If the fan is mounted on a suspended floor and/or it is running at a low speed then stability can prove a problem. If space, or some other reason, precludes the use of an appropriately designed base frame, then an inertia base can be used. The inertia base would be designed to lower the centre of gravity of the fan equipment and incorporate isolator mounting points to give even loading. As a rough guide, the inertia base should be about three times as heavy as the fan equipment to which it is attached.

There are numerous advantages which accrue from the use of an inertia base. The more important ones are listed below:

(i) Equipment stability is enhanced on two counts — movements due to both torque and pressure reaction are reduced.

(ii) Vibration amplitude is reduced.

(iii) Errors in the location of the centre of gravity and the consequent isolator height variations minimised.

(iv) Noise radiated below the fan is reduced.

(v) Coupling between period forces, which may arise from aerodynamic sources, rotor out of balance or uneven isolator loading is reduced.

7.8.4. When a fan is mounted with its shaft axis vertical the most important vibratory force, rotor out of balance, is in the horizontal plane and not, as is more usual, in the vertical plane. It then becomes important to select a mount which has a lateral stiffness, and therefore natural frequency, lower than these properties in the vertical direction. If this is adhered to then the normal isolator selection procedure should give a satisfactory result provided the mount has equal lateral stiffness in all directions.

Whilst most types of mount have a lateral stiffness, which is less than the vertical stiffness, the amount of movement in the lateral plane is usually restricted for stability reasons. There are, however, isolator designs with the desired characteristics which, typically, are illustrated in Fig 7.15.

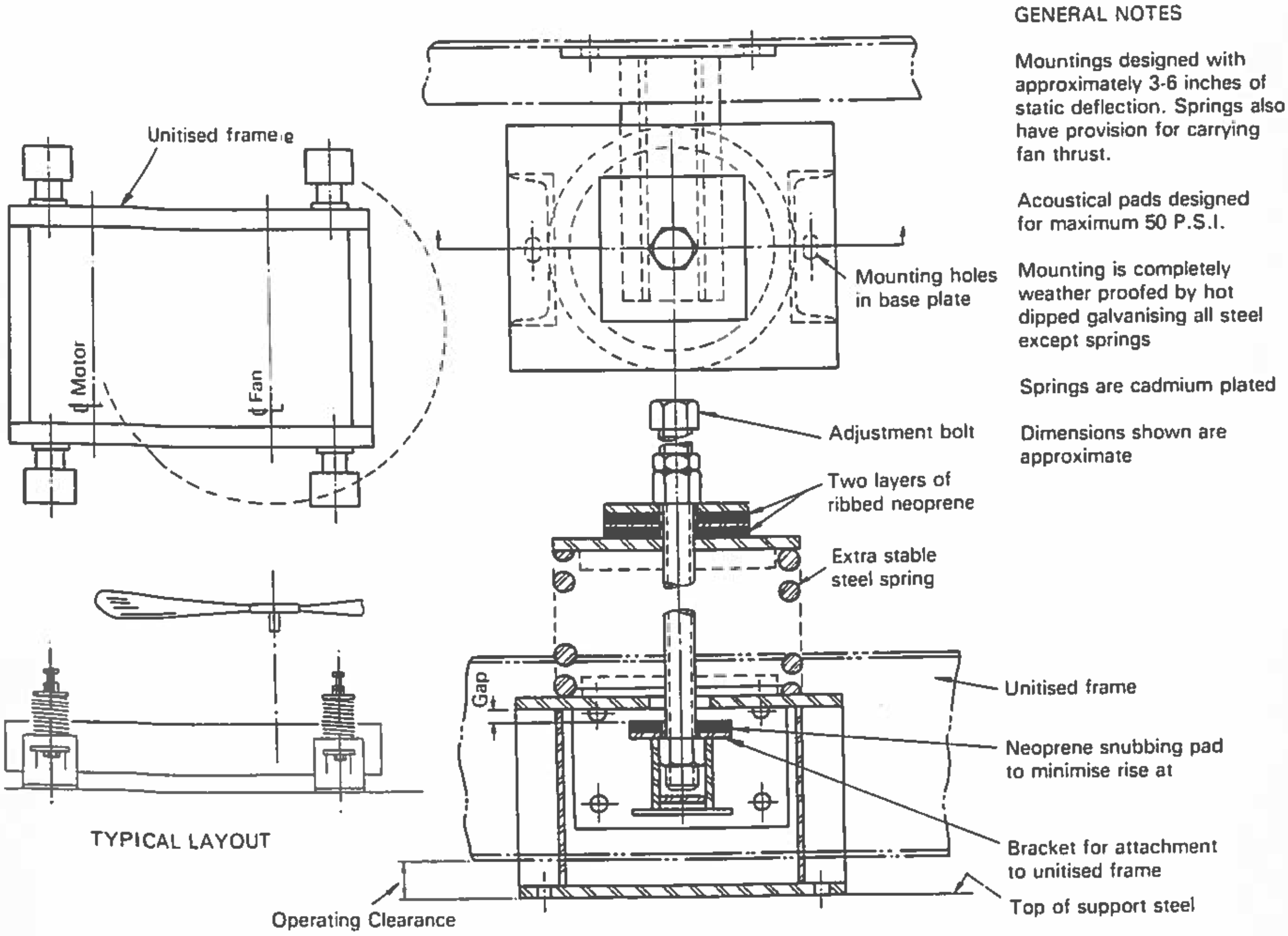


Fig. 7.15. Vibration Isolators with Restricted Lateral Movement

In order to avoid coupling of the out-of-balance forces and a potential pitching motion, if the line of action of the horizontal exciting forces is well away from the support plane, it is usual to mount the fan in two planes. If this is not possible then the effect of any pitching can be minimised by 'spreading' the mounting points away from the fan.

7.8.5. Though perhaps surprising, one of the most common faults in a supposedly isolated system is mechanical 'bridging'. Most fan installations have to be joined to ductwork, either upstream or downstream or both, and an electrical supply has to be connected.

Where required, ductwork must be joined to the fan via a flexible connector. This connector must be flexible enough to avoid adding any stiffness to the system and therefore transmitting vibration into the ductwork.

Electrical conduit should also be provided with a vibration break or joined to the fan with sufficient flexibility to avoid transmission of vibration or noise.

An example of how the former feature can be provided is given in Fig 7.16.

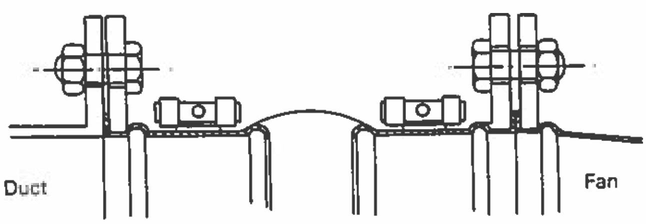


Fig. 7.16. Flexible Connector

8. SITE PROBLEMS INSTRUMENTATION & MEASUREMENT

8.1. Sound Level Meter

The basic sound level meter consists essentially of a microphone to convert sound into an electrical signal; an amplifier to boost that signal; a weighting network to adjust the gain of the amplifier so that it amplifies some parts of the audio spectrum more than others and, at the output of the amplifier, a meter of some description to display the adjusted output voltage. The output voltage is thus related in a defined way to the input sound level, so that the output meter can be calibrated directly in sound units rather than volts.

Some of the less expensive sound level meters consist of little more than these basic components. They are usually fairly inaccurate, have a fixed weighting network, no analytical facilities and are usually referred to by the manufacturer as "Industrial Grade (BS 3489)". At present, the only other 'official' grade of instrument available, and the one that is generally recommended for serious investigative work, is the "Precision Grade (BS 4197)".

The block diagram of a typical good quality portable sound level meter looks like this:

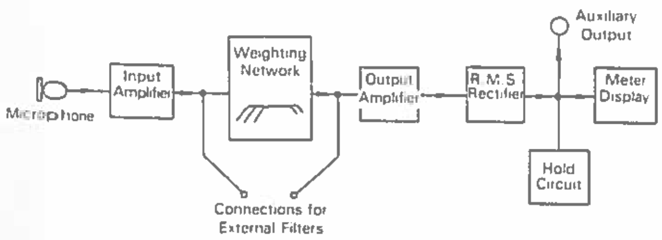


Fig. 8.1.

8.2. Microphone

Microphones convert a sound into an electrical signal which is an analogue of that sound *as heard by the microphone.*

It is important to appreciate that the directional characteristics of microphones (i.e., the extent to which they respond more to sounds coming from one direction than from another) vary from type to type. No microphone responds in exactly the same way as a pair of human ears.

When a microphone is placed in a sound field it disturbs and alters the field in the immediate vicinity of the microphone diaphragm.

This effect is more pronounced with 1 inch diameter microphones than the smaller (½ inch) ones now in common use. The disturbance affects sounds of a higher frequency than about 3-4 kHz and, although no alteration can be detected by the human ear, quite significant errors can be introduced if the necessary precautions are not taken.

There are three main types of microphone for use in conjunction with sound level meters.

8.2.1. **The Free Field Microphone** is calibrated to compensate for its own disturbing presence in the sound field *when pointed directly at the sound source.*

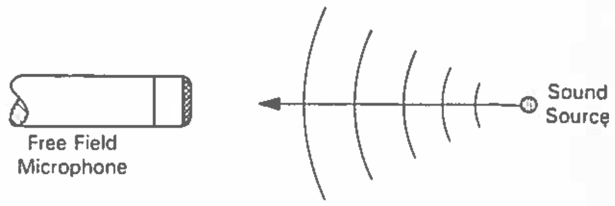


Fig. 8.2.

If it is not pointed at the source, or if the sound is coming from more than one direction, the calibration of the instrument, which takes account of the directional sensitivity of the microphone as well as its disturbance effect on the sound field, no longer holds good.

8.2.2. **The Pressure Microphone** is not calibrated to compensate for its own disturbing presence, but the magnitude of the effect is reduced to negligible proportions if the microphone is pointed not at the sound source but *at an angle of 90° to it.*

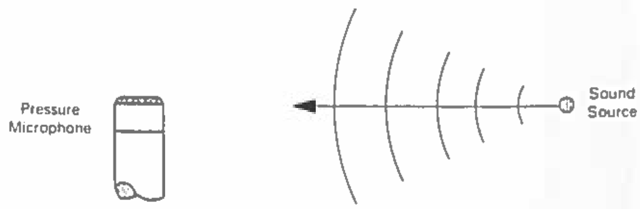


Fig. 8.3.

There are a few very specialised applications where the use of a pressure microphone is to be preferred to a free field microphone, but not many. Most stem from the fact that the larger the microphone, the greater its disturbance effect. When very low level sounds are to be measured, the greater sensitivity of a large diameter microphone may be needed and since the disturbance effect of a correctly angled pressure microphone can almost always be ignored for practical purposes, a large pressure microphone may be the better tool for the job.

8.2.3. **The Random Incidence (or Omnidirectional) Microphone** is designed to have an equal response, and hence uniform accuracy, for sound arriving simultaneously from all directions.

The true random incidence microphone, or a free field or pressure microphone specially adapted to display omnidirectional characteristics, is by far and away the most common and useful instrument in service today. Its use is essential when measuring sound in a reverberant location or when the sound source is very large or diffuse.

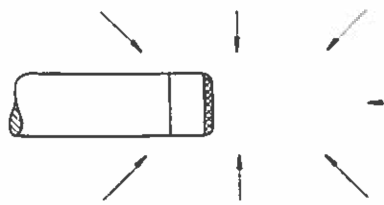


Fig. 8.4.

In general, the smaller the microphone the better its omnidirectional characteristics, but the lower its sensitivity. This lack of sensitivity can create difficulties when attempting to measure low level sources. However, most manufacturers offer a device they call a 'random incidence corrector' which can be fitted over the top of their larger, more sensitive, microphones to give them excellent omnidirectional characteristics and so overcome the sensitivity problem.

A microphone with good random incidence characteristics is undoubtedly the best choice for virtually every measuring situation encountered on site, and, for all practical purposes, its disturbance effects can be ignored.

8.3. Weighting Networks

A weighting network is an electrical circuit which deliberately distorts the signal to simulate the response of the human ear and provide a single figure readout which is a fair representation of the loudness of the sound as perceived by the average listener. The simpler sound level meters will give an 'A' weighted readout only, but the better models have switchable networks offering the user a choice of A, B, C, D or linear (undistorted) weighted displays.

8.4. Filter Networks

Filter networks allow measurement of sound levels within specific bands of frequencies. For most purposes it is sufficient to measure levels in bands one octave wide, but occasionally narrower bands must be analysed. Filter networks are therefore generally available with one octave or one-third octave bandwidths, switchable to allow selection of the appropriate band of frequencies. They come either as an integral part of the sound level meter or as an add-on extra.

Very occasionally, there is a need to examine an extremely narrow band of frequencies, or even a single discrete frequency. Narrow bandwidth tunable filters allow this to be done, but as these devices are both seldom needed and very expensive (even by acoustic measuring instrument standards) they will not often be encountered outside a well-equipped laboratory.

8.5. The Rectifier and Meter Response

The purpose of the meter rectifier circuit is to feed the sound level meter's display device with an electrical analogue of the Sound Pressure Level being measured, after it has been weighted or filtered. Usually the circuit provides a choice of meter response, or damping, characteristics: i.e., **'fast'** and **'slow'**. With the response set to 'fast', the display device will follow (or attempt to follow) any fluctuations that may occur in the Sound Pressure Level. If these are rapid and repetitive it will be quite impossible to obtain accurate readings and, due to statistical effects, such inaccuracies can be greatly magnified when taking measurements of low frequency sounds.

The 'slow' response electronically averages out these fast fluctuations and gives a more or less true average reading; or at least one that is far more accurate than could otherwise be obtained.

The more expensive instruments often give a choice of two further types of response: *'Peak'*, which gives a true reading of the maximum level reached by a sound of very short duration, and *'Impulse'*, which gives a similar reading but weights it to take account of the subjective effect of short duration sounds on a listener (the shorter the duration, the less the apparent loudness). Facilities are always provided to 'hold' these readings for long enough for them to be noted by the observer.

8.6. Calibrators and Calibration

As with any measuring instrument, the sound level meter must be regularly calibrated. Methods employing reference signals generated within the instrument itself are not particularly satisfactory and external reference sound sources such as piston phones, falling balls or electronically generated sounds are preferable.

8.7. Sundry Equipment

8.7.1. **Microphone windshield.** As even low velocity air movement around a microphone can generate noise which will result in inaccurate measurements, the use of a windshield is advisable, however, a windshield can provide only minimal protection.

8.7.2. **Tripods and microphone extension rods or cables** are important when taking measurements in a diffuse or reverberant field where an operator standing close to the microphone would block sound arriving from some directions.

8.7.3. **Headphones** when used to supplement the meter reading can be of help in identifying a sound and thus its source and cause. It is important that the electrical impedance of the headphones be correctly matched to the output impedance of the meter.

8.7.4. **Tape recorders,** which must be of 'professional' standard, are necessary when laboratory analysis of a noise is required. They are also used for analysis of short duration sounds and are particularly useful when an accurate determination of reverberation time is to be obtained by measuring the decay time of an impulse noise such as a pistol shot.

8.7.5. **Calibrated sound sources,** for calibrating the environment, not the meter, can be used for direct measurement of the sound power level of a source by substitution or juxta-position. One common use is the determination of the insertion losses of acoustic attenuators.

8.8. The Vibration Meter

Although there are many specialised vibration meters available, the vibration meter is basically so similar to the sound level meter that most manufacturers of good quality sound meters offer a range of accessories which allow their instruments to be used for measurement of both sound and vibration. The main alterations to the meter when it is used for measuring vibration are that the microphone is replaced with a wide frequency range vibration pick-up (transducer) and the readout display is calibrated in units of vibration rather than sound.

8.9. Vibration Transducers

Although transducers are available to measure displacement, velocity or acceleration, the piezoelectric accelerometer is by far the most commonly used. It is small, tough, stable and reliable, can be mounted in any attitude, is relatively cheap, easy to calibrate, has no wearing parts and has an excellent combination of a wide frequency and large dynamic range. The only minor disadvantage is that its electrical output impedance is high and consequently the electrical output signal must be fed through a special preamplifier before it can be processed by the meter. Because velocity is the first integral of acceleration, and displacement its second integral, the inclusion of a simple electronic integrating circuit enables the meter to show direct readings of all three parameters. Purpose-designed vibration meters invariably include built-in switchable integration circuits whereas adapted sound level meters use a separate integration unit which has to be connected between the accelerometer and its preamplifier.

Measurement

8.10. Measuring position

When checking the Sound Pressure Level in a room or treated space for compliance with a specification designed to set a limit on general noise, or when trying to evaluate the severity of a "noise problem", the microphone is set-up where a human ear, in going about its normal lawful business, is likely to be most affected! However, when investigating noise from a given fan source the Sound Pressure Levels in the vicinity of the fan must be measured in such a way as to allow calculation from them of the Sound Power Level produced by the fan itself.

In the event that the fan Sound Power exceeds its specification then the fan, its associated motor and drives, and its installation must be examined. If, on the other hand, it is established that the fan Sound Power is within specification but the measured Sound Pressure Levels are too high, then it is in the system and/or the environment where the problem lies. Either way, the fan Sound Power must be determined with a reasonable degree of accuracy and this can only be done if the measuring position is chosen with care.

The area in which measurements must be taken is invariably part anechoic and part reverberant — never one or the other; and theoretically this area can be divided into three distinct zones called the *Near Field,* the *Free Field* and the *Reverberant Field.*

In the Near Field, which for most purposes can be taken to extend for something like 11ft from the fan (to be precise, it is the wave length of the lowest frequency to be measured or twice the fan diameter, whichever is the greater), the Sound Pressure Level varies enormously with very small changes in measuring position.

Within the Free Field, sound propagation is subject to the inverse square law so that doubling the distance between the microphone and fan will decrease the Sound Pressure by 6dB. Readings should be made in this zone if possible. Beyond this region is the Reverberant Field where reflected sound, from walls and other objects, becomes of similar magnitude to the direct sound from the fan.

Very often there will be no Free Field in the area in which measurements must be made either because the room is too small or because it is too reverberant. In these circumstances the reverberation time must be measured or estimated in order to convert the measured Sound Pressure Levels to the fan Sound Power Level.

A useful little table, found in a number of textbooks on acoustics, is reproduced below. It is perfectly adequate for most site investigations and will often avoid the need for a time consuming analysis of a pistol shot recording to determine a reverberation rating.

8.11. Typical Reverberation Ratings

Use of Room	Sound Absorbing Surfaces	Reverberation Time, Seconds	Rating
Radio and TV Studios, Music Rooms	Special ceiling, many people, and soft furnishings	0.2 to 0.25	Dead
Department Stores Restaurants	Acoustical ceiling, many people, and soft furnishings	0.4 to 0.5	Medium Dead
Offices, Libraries Homes, Hotel Rooms, Conference Rooms, Lecture Halls	Acoustical ceiling or soft furnishings), many people	0.9 to 1.1	Average
School Rooms, Art Galleries, Building Lobbies, Hospitals Small Churches	Many people or some acoustical material	1.8 to 2.2.	Medium Live
Large Churches, Gymnasia, Factories	People only	2.1 to 4.5.	Live

There will be occasions when it is immediately obvious that the problem lies not with the fan but with the environment. In such cases it is often worthwhile assuming the fan Sound Power to be as specified and calculating a theoretical Sound Pressure Level for the room or area working from this basis. A comparison of the predicted and measured Sound Pressure Levels will sometimes quickly pinpoint the source of the trouble.

8.12. Influence of Operator on SPL Readings

One's body and even the casing of the meter can block out sound coming from certain directions and thus prevent it reaching the microphone.

Both can also cause further measuring inaccuracies by reflecting sound on to the microphone. One authority quotes errors of up to 6dB in the 400Hz region when the microphone is held at less than 1 metre from the operator's body.

Most sound level meters are designed in such a way that case reflections are minimal and it is usually sufficient to hold the microphone at arm's length. However, if there is any doubt, a microphone extension rod or tripod with an extension cable must be used. The effect of one's body on a reading can then be seen by stepping to one side whilst maintaining the microphone in a fixed position.

8.13. Background Noise

Background noise is always present on site and may sometimes make measurement impossible. Where the difference with the fan running and the fan off is less than 3dB, accurate measurement is not possible. Between 3dB and 10dB difference a correction factor must be subtracted from the fan + background measurement to obtain the level attributable to the fan itself. With a difference greater than 10dB, no correction is necessary.

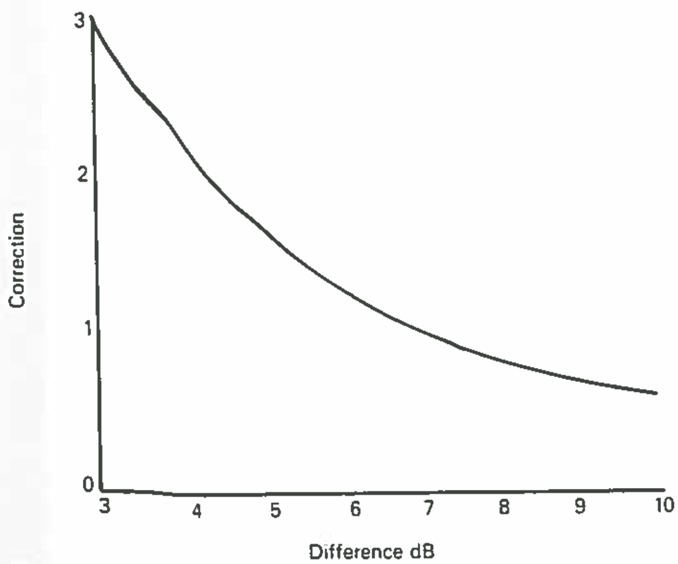


Fig. 8.5.

Unless vibration levels have been specified and are to be checked, or a vibration problem has been reported, vibration measurements would normally only be undertaken as an aid to tracing the cause of a previously observed excessive Sound Level.

8.14. Accelerometer Mounting

The accelerometer must not alter the character or frequency of the vibration being measured. Fortunately, the mass of a piezoelectric accelerometer is usually small in relation to the mass of the vibrating object being investigated and its effect on the vibration is negligible. Errors, however, can sometimes be introduced in this way so this must always be borne in mind.

Accelerometer manufacturers always offer a variety of mounting methods and attachments for their products. If a mounting difficulty is encountered it is well worth looking at the manufacturers' literature to see what is available, although in practice one of the following three methods can usually be employed: double-sided adhesive tape for non-magnetic components; magnetic bases for most steel components and a hand-held probe. Double-sided tape or a magnet are to be preferred when an actual measurement is to be taken.

Although the hand-held probe can be extremely useful for *tracing* a vibration, it is virtually uselessfor obtaining an *absolute* measurement because pressing it against a component usually reduces the amplitude of the vibration; in some cases stopping it altogether (e.g., ductwork panels). Its upper frequency limit is about 1000Hz, but this does cover most of the vibrations needing investigation.

When taking readings with a piezoelectric accelerometer the connecting cable must not be in contact with a vibrating surface and must not be allowed to 'whip'. This is because the high output impedance of the accelerometer makes the cable vulnerable to induced disturbing signals (known as 'microphony') caused by cable vibration, which will result in measurement inaccuracies.

8.15. Investigation Reports

Careful documentation of site measurements and conditions is essential, particularly when an off-site analysis may be necessary or when comparisons are to be made at some future date.

A good investigation report should contain at least the following information irrespective of whether or not it may seem relevant or obvious at the time because others without background knowledge of the situation may need to refer to the report:

(i) Instrumentation used — makes, types and serial numbers.
(ii) Dates and methods of calibration.
(iii) Weighting networks and meter responses used.
(iv) Nature of sound or vibration (e.g., random, continuous, any pure tones, etc.).
(v) Background noise levels.
(vi) All available data on object investigated (e.g., fan type, speed, blade pitch, etc.).
(vii) Sketch of measuring location with all relevant dimensions; locations of microphone, accelerometer, other objects; details of absorbent or reflective surfaces, etc.

8.16. A few basic Rules (Technical)

Before visiting a site:

(i) Obtain as much information about the fan, site and installation as possible. In particular, obtain details of the theoretical or previously measured sound characteristics of that type of fan.

(ii) Read the instruction books and become thoroughly familiar with the operation of the measuring instruments.

(iii) Check the instrument batteries and take a spare set.

(iv) Check that the instruments have recently been properly and appropriately calibrated for the type of measurement required.

Once on site:

(i) Take 'orientation' measurements before starting to record values for the report in order to get a 'feel' of the site. For instance, it will help to determine the type of sound field, whether or not there are any low frequency standing wave patterns and so on, all useful when deciding the best way of setting about the investigation.

(ii) If extraneous noise is causing confusing sound level meter deflections headphones can help identify the source.

(iii) Whilst taking noise measurements:

(a) Avoid absorbing or reflecting surfaces. Ensure that one's own body cannot affect the readings.

(b) Try to take measurements at the optimum distance from the sound source.

(c) Avoid siting the microphone behind objects.

(d) Use a microphone windshield, even in low velocity airstreams.

(e) Set the meter range so that meter readings are as near as possible in the centre of the range. Distrust readings falling in the bottom 10% and top 3% of the range. Never accept overloaded or overdeflected meter readings.

(f) Always remember to measure and record the background noise levels even though they may appear to be negligible.

(iv) Prepare and keep a well documented investigation report.

8.17. A few more basic rules ('others')

'Other' considerations besides purely technical ones influence site investigations. There are rarely included in the standard works on acoustic measurements, perhaps because the authors consider them too frivolous to mention. However, because ignoring them will inevitably result in the loss of someone's time, money or temper, here are a few:

(i) Dress is important.

Try to establish in advance the type of site to be visited and, if building is still in progress, the stage of completion reached.

Arriving dressed in an office suit at a site where the only access is via a quagmire will be inconvenient to say the least.

On the other hand, it is an inexplicable but indisputable fact that the appearance in an occupied building of someone dressed in 'construction site' clothing, carrying a sound level meter, has the immediate effect of substantially increasing subjective noise levels!

(ii) Commissioning of equipment usually takes place before a construction site is cleared and loud intermittent noises and vibrations are an inherent feature of construction work.

So, to avoid the need to politely ask nearby machinery operators to "hang on a bit", a request to which they seldom take kindly, it is worthwhile establishing the opening hours of the 'local' and arranging measuring times accordingly.

(iii) Unless one makes careful enquiries beforehand, *expect* to arrive at an occupied block of flats at 10 a.m., say, only to be informed that "It's only noisy when we're in bed at night".

(iv) When investigating a complaint originating from an occupant of a building, particularly if it is his or her home, always try to meet the complainant in person. Irrespective of whether or not the complaint is justified, or whether or not the noise can be reduced, a friendly chat can very often stop further complaints being received from a lonely O.A.P.

(v) "Could you tell me where the fan is, please? I need to take some sound measurements."

"You can't. It was disconnected a week ago because it was too noisy!"

(vi) A consultant demands an immediate visit to a site 300 miles from one's office — he was there yesterday and the fans sound like jet aircraft taking off!

One arrives on site, finds that none of the fans has yet been connected to the electricity supply, and watch a Boeing 747 taking off from the nearby airport!

(vii) One will often find that the removal of transit packing pieces from a noisy unit will save a considerable amount of unnecessary investigation.

(viii) Be prepared for the fact that typical measuring locations include:

(a) Exposed roofs, 11 stories high, whipped by Gale Force 8 winds.

(b) Boiler rooms, seemingly never exceeding 6ft x 8 ft in area, equipped with an array of badly lagged pipes all with tremendous potential for causing second degree burns.

(c) Ceilings or under-roof areas where access must be gained by a 30ft ladder with four rungs missing and the only available support is a lighting track.

9. CONDITION MONITORING

9.1. Vibration Identification

In the chapters so far, noise and vibration have been considered in a general way and permissible overall limits have been indicated. For important and/or arduous applications, however, where a total breakdown could be catastrophic, the causes and effects of vibrations must be identified.

The identification of the cause of a vibration is to be found in an analysis of its frequency and velocity. The amplitude of vibration is not important in a diagnostic sense except when considering frequencies below about 10 Hz.

Vibrations occur at different frequencies depending on their cause, each type of cause being identified by a specific frequency (or multiple of that frequency) in a manner similar to a human being uniquely identified by a fingerprint. For example, a faulty ball bearing causes high frequency vibration at many times the fan frequency, whilst unbalance or misalignment produces vibration at the fan frequency itself.

Problems, and, importantly, developing problems, cannot be identified by measuring the vibration amplitude. Furthermore, the acceptable amplitude is related to the frequency — the lower the frequency the greater is the acceptable amplitude (CHAPTER 6) *BUT* the vibration *VELOCITY* should remain constant.

The 'vibration signature' of the fan set must be obtained immediately after successful commissioning. This is the 'standard' against which subsequent vibration analyses are compared.

The 'signature' is obtained by holding or attaching a transducer (CHAPTER 8) to certain parts of the installation, usually the fan and/or motor bearings, and viewing the display on the associated meter, Fig 9.1. To produce a permanent record of the analysis, an X-Y recorder, connected through a tunable frequency analyser, is used to plot the vibration velocity against the narrow frequency bands which the analyser can identify, Fig 9.2. The vibration transducer can be attached to the bearings to measure the vertical, horizontal or axial modes. The resultant traces provide more detailed information about possible sources of vibration.

Subsequent monitoring will highlight any increases in vibration velocities, and these increases will warn of the onset of problems. The frequencies at which the increases have occurred will identify the problems.

Vibration velocities are a quality judgment whilst their frequencies will indicate the cause.

9.2. Vibration Sources & Their Frequency 'Signatures'

9.2. 1.The Fan in General

(i) UNBALANCE — the most common cause of vibration. As a heavy spot gives a 'pulse' to the transducer once every revolution, unbalance in a fan is identified by high readings in the horizontal and vertical directions at the rotational frequency, Fan $\frac{rpm}{60}$ Hz.

High readings at commissioning can indicate residual unbalance in manufacture, or 'sag' if the fan has not been rotated regularly during storage on site. The build up of dust on an impeller, erosion or corrosion also leads to increasing vibration. Scaling at high temperatures (above 400°C) or soaking in heat whilst stationary are other possible causes of unbalance.

(ii) MISALIGNMENT — almost as common as unbalance despite the use of self aligning bearings and flexible couplings. Fig 9.3. illustrates possible types of coupling misalignment. A bent shaft also produces misalignment.

Misalignment always produces radial *and* axial forces, the size of these forces, and therefore the vibration produced, being proportional to the misalignment. The axial readings are usually 50%, or more, of the radial readings and again the frequency is normally $\frac{rpm}{60}$ Hz.

When the misalignment is severe however, vibration at $2 \times \frac{rpm}{60}$ Hz and even $3 \times \frac{rpm}{60}$ Hz may be experienced.

Misalignment can also occur when bearings are not correctly lined up or, very often, when a fan has been distorted by tightening down on to foundations which themselves are not level. With sleeve bearings this will produce vibration proportional to the residual unbalance, but with ball or roller bearings an axial vibration would be produced even if the impeller were 'perfectly' balanced, a condition which is impossible to achieve.

Another very common fault is incorrectly aligned pulleys and Vee belt drives, Fig 9.4. This misalignment causes rapid wear of belts and production of frictional heat which can be transmitted to, and affect, shafts and bearings. It eventually can result in destructive vibration.

High axial readings (at least 50% of the horizontal or vertical) at $\frac{rpm}{60}$ Hz and possibly $2 \times \frac{rpm}{60}$ Hz or even $3 \times \frac{rpm}{60}$ Hz indicate probable misalignment.

(iii) ECCENTRICITY — occurs when the centre of rotation does not coincide with the geometric centre.

As far as a fan impeller is concerned, this normally leads to more mass on one side of the rotational centre than the other, i.e., unbalance. It can therefore be corrected

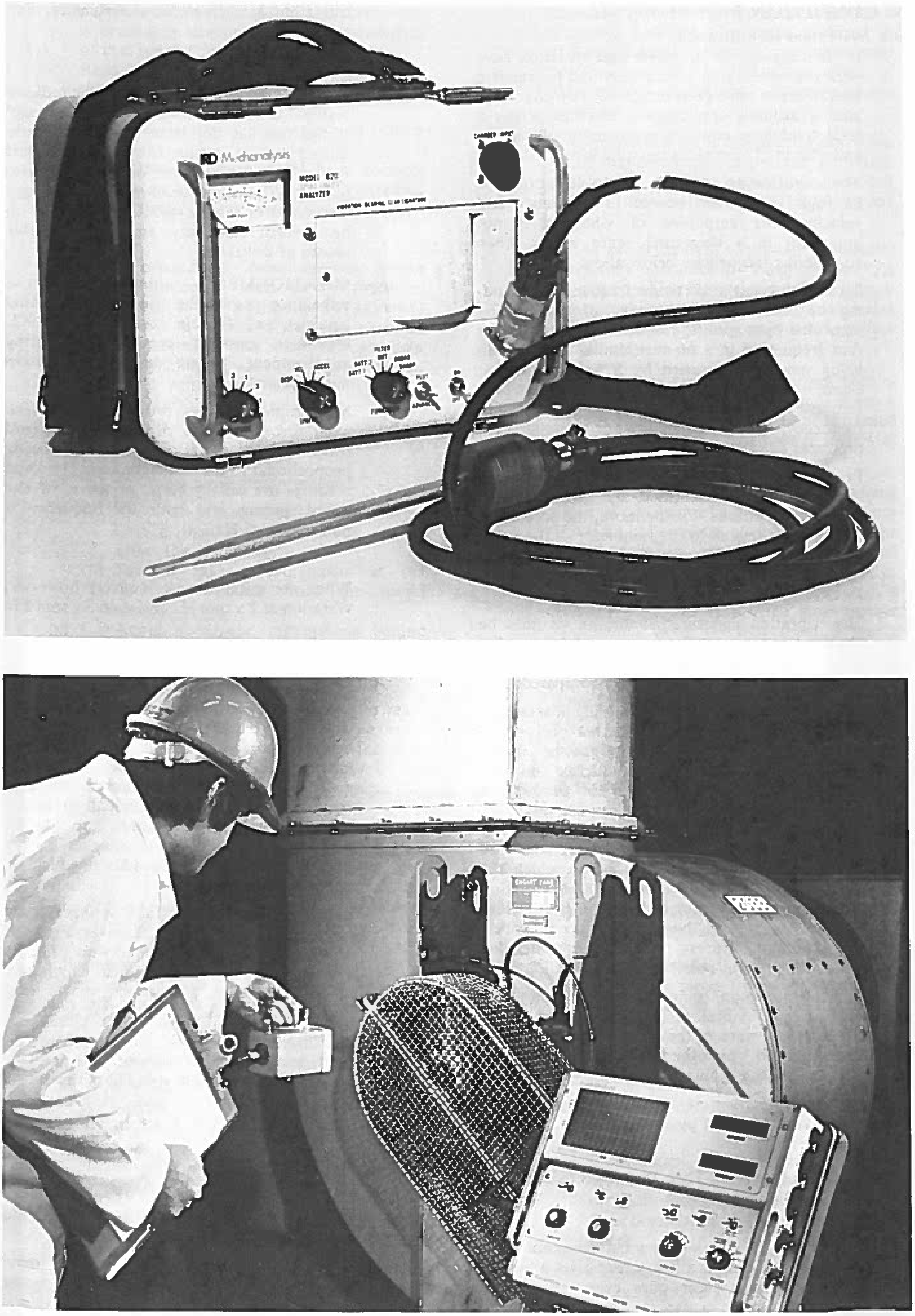


Fig. 9.1. Typical Apparatus

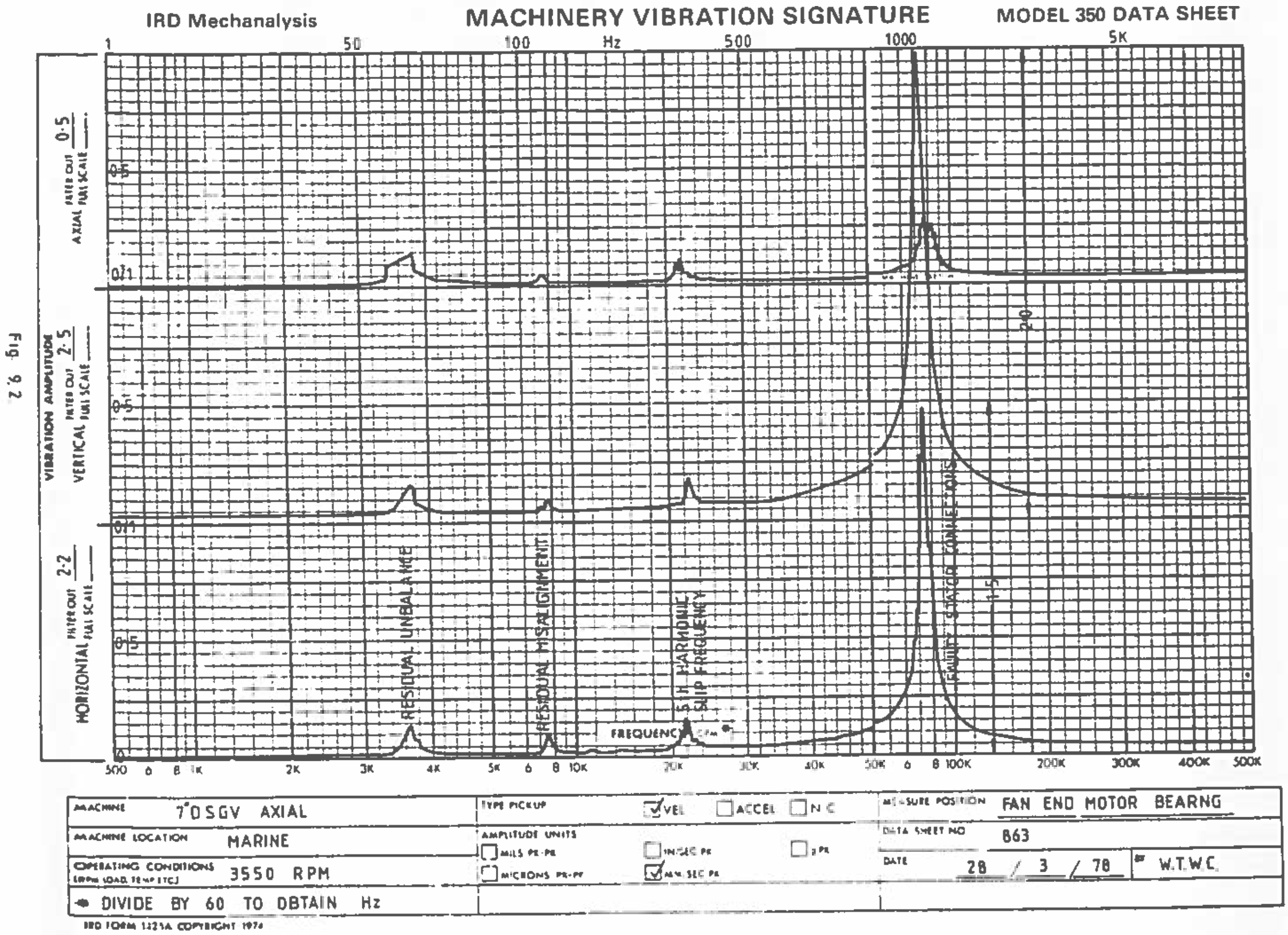


Fig. 9.2.

providing rebalancing is carried out with the impeller on its own shaft and bearings, and the positions of the inner races of ball or roller bearings on the shaft are not changed.

The predominant vibration frequency is, of course, $\frac{rpm}{60}$ Hz.

When fans are gear driven, eccentricity can produce reaction forces between the pinions because of the cam-like action. Vibration will be at a maximum along a line joining the pinion centres, at a frequency of $\frac{rpm}{60}$ Hz of the eccentric pinion. The problem cannot be cured by rebalancing.

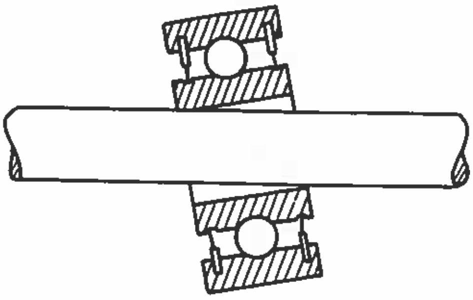
A similar situation can arise with eccentric Vee belt pulleys. Vibration will be at its maximum in the direction of belt tension at a frequency of $\frac{rpm}{60}$ Hz of the eccentric pulley and, again, the problem cannot be cured by rebalancing.

(iv) LOOSENESS — usually resulting from loose foundation bolts or excessive bearing clearances.

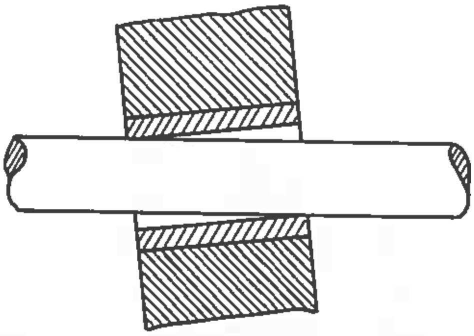
Vibration will not occur unless there is an exciting force such as unbalance or misalignment. However, quite small forces can produce large vibrations. Although rebalancing, or realignment would theoretically eliminate vibration, the extreme accuracy necessary may be impossible to achieve on site and, therefore, it is essential to tackle the problem at its source.

To determine the characteristic frequency of any looseness, consider an unbalanced impeller fitted to a shaft running in a bearing with loose holding-down bolts. When the heavy spot is downward, the bearing will be forced against its pedestal. When the heavy spot is upward, it will lift the bearing. At positions 90° to either side the unbalance force will neither lift nor hold down, and the bearing will drop due to weight alone. Thus there can be two applied forces each revolution of the shaft and the characteristic frequency of looseness could be $2 \times \frac{rpm}{60}$ Hz.

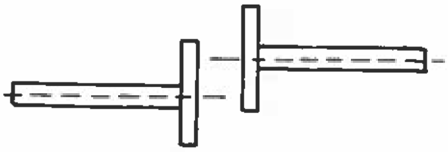
Furthermore, a small rotating force could cause slight movement in any radial direction.



When an anti-friction bearing is misaligned with its shaft, axial vibration will occur whether unbalance is present or not.

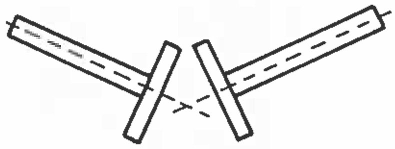


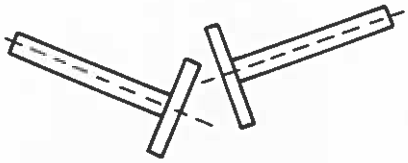
Misalignment of a sleeve-type bearing with its shaft will only cause axial vibration if accompanied by unbalance



Offset Misalignment

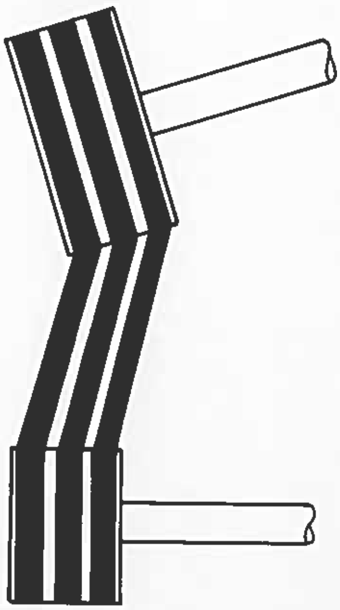
Angular Misalignment





Combination Angular/Offset Misalignment

Fig. 9.3. Misalignment



Angular or offset misalignment of "V" belt pulleys causes high axial vibration.

Fig. 9.4. Vee belt drives

(v) AERODYNAMIC FORCES

These are not usually a problem unless they excite some part of the fan or associated ducting system to vibrate at resonance (see below).

An investigation of the fan noise spectrum will usually indicate possible troublesome frequencies, especially if a narrow band frequency analysis is available. In general terms, however, the following frequencies may be the source of trouble:

a. Blade passing frequency $= \text{No. of blades} \times \frac{\text{rpm}}{60}$ Hz

b. Guide vane frequency (Inlet guide vanes on a centrifugal or axial fan). $= \text{No. of guide vanes} \times \frac{\text{rpm}}{60}$ Hz

c. Support frequency (e.g., motor supports on an axial impeller, tie rods on a centrifugal). $= \text{No. of supports} \times \frac{\text{rpm}}{60}$ Hz

d. Secondary frequencies at the harmonics of the above may also be present.

e. Interactive frequencies will occur at (a + b) Hz, (a - b) Hz, (b + c) Hz, etc.

NOTE: Where the number of blades and guide vanes are equal, both even integers, both odd integers or differ by an even integer, there is the possibility of beat frequencies being set up.

Every object has a natural frequency at which it tends to vibrate. Should an aerodynamic forcing frequency coincide with the natural frequency of a part, then resonance will occur. Overcoming the problem with a Vee belt driven unit is simple because a small change in the rpm will normally be sufficient. With a direct drive unit, stiffening, or a change in the aerodynamic design, may be necessary. The problem is very unlikely to occur in the fan, but is often a problem with associated ductwork. Special care should be taken with the design of inlet and outlet bends on fans as, with an uneven velocity distribution, resonance can occur. The usual solution is to introduce splitter vanes (taking care with their number to obviate beat frequencies) or make bends of larger radius.

9.3. The Specific Problems of Fan Bearings

9.3.1. Sleeve Bearings

Problems generally result from excessive clearance, wiping, erosion of the journal surfaces (caused by builders' dust on site entering the bearings before start up), looseness of the white metal, inadequate lubrication (poor maintenance), lubrication with an incorrect grade of oil, or chemical corrosion. Characteristic vibration frequencies are $1 \left(\frac{\text{rpm}}{60}\right)$ Hz, $2 \left(\frac{\text{rpm}}{60}\right)$ Hz

9.3.2. Ball & Roller Bearings

Races which have flaws on the balls, rollers or raceways will usually cause high frequency vibrations, identified as follows:

Flaw in outer raceway or variation in stiffness around the housing: $\frac{\text{rpm}}{60} \times \frac{b}{2}\left(1 - \frac{h}{H}\cos M\right)$ Hz

Flaw in inner raceway: $\frac{\text{rpm}}{60} \times \frac{b}{2}\left(1 + \frac{h}{H}\cos M\right)$ Hz

Flaw in ball or roller: $\frac{\text{rpm}}{60} \times \frac{H}{h}\left(1 - \frac{h^2}{H}\cos^2 M\right)$ Hz

Irregularity in cage or rough spot on ball/roller: $\frac{\text{rpm}}{60} \times \frac{1}{2}\left(1 - \frac{d}{H}\cos M\right)$ Hz

Where b = number of balls or rollers.
h = diameter of balls or rollers
H = pitch circle diameter of race
M = angle of contact of ball or roller.

These vibrations not being easily transmitted to the fan the flaws will be discovered by velocity readings on the bearing housing.

Severe misalignment or a race will sometimes result in a frequency at $n \times \frac{\text{rpm}}{60}$ Hz even when the bearing itself is satisfactory.

Modern ball and roller bearings are precision made and with correct installation and lubrication are unlikely to cause trouble. When they fail it is more often the fault of unbalance, misalignment, lack of proper lubrication, or use at speeds, loads or temperatures in excess of those recommended by the manufacturers.

When faults are present, vibration readings at the relevant frequencies are usually at least five times the reading produced by a good bearing.

9.4. Specific Vee Belt Drive Problems

Many of the problems found in fans can also be present in Vee belt drives. The balancing of pulleys is often overlooked and must be specified when ordering. Misalignment has already been mentioned.

Vee belt drives have good resistance to shock and vibration, but may be blamed for causing trouble as they can be seen to whip and flutter, especially when the belts are unmatched. Belts are often changed unnecessarily when the fault is really that of unbalance, misalignment, eccentricity or looseness. Nevertheless, the importance of using matched sets of belts cannot be emphasised sufficiently.

Vibration from faults in the belts themselves occur at multiples of belt speed. The relevant frequencies are

$$1, 2, 3 \text{ or } 4 \times \left\{\frac{\text{Pulley diam}}{\text{Belt length}} \times \frac{\text{Pulley rpm}}{60}\right\} \text{Hz}$$

Likely faults in the belts are rough edges due to pieces being broken off, and hard or soft spots.

Faults in pulleys, such as chipped grooves, will be identified at the speed of the relevant pulley $\frac{rpm}{60}$ Hz

9.5. **Electric Motor Problems**

Most electric motor vibrational problems are mechanical in origin usually resulting from unbalance, misalignment, bolting down to foundations which are not level, loose foundation bolts or faulty bearings. Previously listed frequencies are therefore applicable using, of course, the motor rpm where this differs from the fan rpm.

With induction motors forces act in the air gap between rotor and stator tending to pull these together and produce vibration at twice the line frequency Hz. Normally such vibration is small except in two pole motors, but if the air gap varies, or if the tightness of stator laminations or windings in the stator vary, then this vibration will increase considerably. The 2nd and 3rd harmonics may also be important.

Generally slip frequency, (Line frequency-Motor rpm/60)Hz, will not in itself be important as it will be of very low frequency. However, its interaction with higher frequencies can produce pulsations.

If the rotor is severely unbalanced, the high spot will come closer to the stator than other points. As it passes the stator poles more pull is exerted. Thus vibration occurs at twice slip frequency on a 2 pole motor, quadruple slip frequency on a 4 pole motor and so on. The magnitude of the readings in these frequencies can indicate whether the problem is simply due to lack of balance, change in the air gap, worn journals, broken rotor bars, etc.

If a resonance condition exists within the motor at line frequency, then large vibrations can be produced. More often however this is the fault of an unbalanced magnetic pull and can be cured by changing stator connections.

With all suspected electrical sources of vibration, the simple check is to switch off the motor when they should "die".

The perfect fan and motor do not exist. It is therefore the *increase* in readings which will indicate trouble.

Vibrational monitoring and diagnosis are the province of skilled persons and the major fan companies have devoted much time to the relevant research. The use of vibration meters with frequency analysis is to be encouraged provided that operators are trained in their use.

Whilst one should approach the manufacturer for advice, it is nevertheless of value to be able to appreciate possible trouble sources in the system. The contractor and user should be aware of how they can ensure and monitor good installation and maintenance so that the inherent good running sought for in design and manufacture continues in service.

10. CASE STUDIES

The cases in this chapter illustrate some of the problems which occur on site and narrate how the installations were improved.

No attempt has been made to make the examples exhaustive — that would have been impossible; but they have been chosen to include a representative selection of fan types and sizes. The *real purpose* of the chapter is to provide an *appreciation* of the range of noise and vibration problems which occur and to show how an analytical approach can identify the cause of a problem and suggest a possible cure.

Case Study 1

Excessive fan noise from a filtered fresh air input system created a problem in two adjacent laboratories of an electronics factory. Although little noise was emanating directly from the unit casing containing a 24″ diameter axial flow fan running at 1,440 rpm, there was considerable breakout from the 600mm wide rectangular ductwork immediately downstream from a 90° radiused bend below the unit. Noise with a large high frequency component was unpleasantly loud in the neighbouring laboratory served by the air input grilles at the discharge end of the branched duct system.

A theoretical analysis predicted noise breakout levels from the system, yet on-site octave band Sound Power Level readings showed noise levels well below those expected in the 125Hz and 250Hz bands and greatly in excess of those anticipated at higher frequencies. The ductwork was well constructed with no excessive drumming or vibration present in any panels and the fan was in perfect running order.

Three possible contributing sources of the excessive noise were observed. Removing the discharge grilles indicated that they were causing the generation of excessive air noise, some of which was transmitted back down the ductwork. Turbulent airflow arising from the 90° bend below the fan and from the regulating dampers at the duct branches gave rise to some regenerated noise. There was also some evidence of regeneration at 500Hz as the duct width was approximately equal to the wave-length at this frequency. A different pattern of discharge grilles and a 90° splitter silencer bend eliminated the problem.

The fan is too often blamed if the system is too noisy.

1. *Grilles are often chosen because of their appearance rather than their suitability.*
2. *Bad system design invariably causes turbulence which, in turn, causes noise.*
3. *Rectangular ductwork, unless acoustically insulated, should be avoided in a conditioned area.*

Case Study 2

A forward curved centrifugal blower in a plant room mounted extract unit was being blamed for undue noise in the bathroom and toilet areas of two adjoining Local Authority flats. Although the specification called for sound levels not exceeding

NC35, the acoustic performance of the fan had not been considered at design stage.

It was observed that the wall panels on which the extract grilles were mounted were excited into vibration by the resonating extract system ductwork. A few extra securing screws stopped the panel vibration and brought the octave band Sound Pressure Level readings down to NC35 levels in all but the 125Hz band. Flexible connections fitted between the fan unit and ductwork brought the 125Hz noise down to an acceptable level, incidentally facilitating easier access for future fan maintenance.

Common faults too often encountered on site are the omission of flexible connectors between fans and ductwork, and badly secured panels or fixings to which the grilles and sometimes fans themselves are fitted.

Case Study 3

A centrifugal blower was blamed for causing distinct noise peaks at approximately 100Hz and 1000Hz in a kitchen extract system. However, Sound Pressure Level readings taken within the large fan housing showed no peaks at these frequencies and it was decided to use a magnetically attached accelerometer to investigate the system further.

Areas of ductwork with a natural resonance frequency at 100Hz were soon located, and silenced by tightening the associated support clamps. This did not affect the 1000Hz noise peaks present in the vertical duct risers of the system, which seemed to be acting like organ pipes. The subsequent site report recommended substituting radiused bends for the sharp right-angled take-offs serving the vertical risers. This work was carried out and a cure effected.

Another example of the fan being incorrectly blamed for producing excessive noise!

Radiused bends should always be used with right angled take-offs.

Case Study 4

Turbulent air flow had caused considerable buffeting and shaking of the casing of a roof extract unit containing a large mixed flow fan. The fitting of special absorbers and air straightening vanes alleviated the problem, but the site engineers reported their concern about low frequency vibration still present in the roof area surrounding the plant.

A vibration level meter was used to check the machine and, although vibration levels were within specification, it was demonstrated that the particularly resonant roof structure was being excited via the fan unit's base. A set of "Double U" shear pattern rubber anti-vibration mountings was selected to complement the compressed-cork type absorbers already fitted and this overcame the problem

An interesting problem initiated by an incorrect system resistance value causing the fan unit to operate in a previously unknown region of instability.

Be careful when lightweight roofs are used to support rotating or vibrating equipment!

Case Study 5

A lecture room at a school for the deaf was served by a warm air input unit and a roof extract unit, both incorporating 12" diameter, 4 pole, axial flow fans. Strange though it may seem, unacceptable noise levels were reported and a theoretical acoustic analysis of the system was carried out. This suggested that NC60 sound levels were being reached and this was confirmed by measured sound levels of NC55+4. It was discovered that special equipment, used in the teaching of the deaf and partially deaf children, was so sensitive that noise levels in the areas where they were being taught had to be kept as low as possible. Discussion with the Local Authority resulted in the recommendation that speed controllable 6 pole fans of higher blade angle be substituted. At full speed the reduction in air volume of about 7% was acceptable in the light of the corresponding 6dB decrease in Sound Pressure Level; while when the special hearing aid equipment was used the fan speed control would enable noise levels to be reduced by a further 22dB yet still maintaining an acceptable air change rate.

A problem caused by lack of information at the design stage. The solution was acceptable and inexpensive in this case, but reducing the duty is often a poor compromise.

Case Study 6

Visits were made to each of two child assessment units run by a provincial Area Health Authority. Two belt-driven backward-curved centrifugal-fan roof extract units were utilised on each building in association with curb mounted sound attenuators, as recommended at design stage following acoustic analysis of the proposed installations. The buildings were not yet occupied and the absence of furnishings, carpeting, etc., made a re-analysis necessary before it could be checked that the fans were within specification.

Octave band sound pressure levels were measured inside the buildings and were in excess of what could be reasonably expected in the 125Hz, 250Hz and 500Hz bands. Examination of the fan units revealed that transit packing pieces had not been removed from the anti-vibration mountings, and one unit had a particularly noisy motor. With these faults rectified, further octave band Sound Pressure Level readings were taken and showed that noise in the 500Hz band was now within the required NC40 level but levels at 125Hz were from 7dB to 10dB in excess of those specified.

Results of laboratory tests indicated that a reduction of 7dB would be achieved at 125Hz and 250Hz by replacing the roller bearing assemblies fitted to the fans with sleeve bearings. This course of action was specified.

It was apparent that some ceiling resonance was contributing to the overall low frequency noise levels and measurements taken with an accelerometer indicated that a further 4-5dB reduction could be obtained by fitting rubber A.V. strips between the units and their upstands. The units were so treated and the system noise levels brought within the specification criteria.

Everyone at fault!

1. *Always check that transit packing pieces are removed by the installer.*
2. *The manufacturer now always uses sleeve bearings on fans running below a certain speed.*
3. *Lightweight building systems are often a source of vibration and resonance.*

Case Study 7

An air extract system using four propeller fan units was installed on the roof of the main hall of a large public school. Noise from the fans was reported as being "unbearable". A theoretical analysis was made of the system using the fan manufacturer's published sound power levels and information available from the builders' work drawings.

On-site octave band Sound Pressure Level measurements were well within the predicted levels in all but the 125Hz octave band and, although it fell outside the practical scope of the analysis, the 63Hz band. Readings were taken as follows:

63Hz	66dB - 70dB	(Background 48dB)
125Hz	64dB - 72dB	(Background 36dB)

Two figures are quoted for each octave band as the noise at these frequencies varied between the levels stated with a slow, steady pulse. Measured levels varied considerably in a regular pattern as the sound level meter was moved from one end of the hall to the other.

The fans themselves were in perfect condition, but the roof construction was such that large areas of decking were readily excited into resonance in the 63Hz and 125Hz octave bands. The hall was exceptionally reverberant at low frequencies and, combined with the resonant roof, encouraged standing wave patterns and resonant low frequency "beats" to manifest themselves when the extract fans were running.

The fitting of coil-spring based A.V. mountings to the fan cases minimised the roof excitation and consequent low frequency reverberant sound pressure levels in the hall.

A very extreme example of the problems being caused by large lightweight decking roofs when they are not locally strengthened where they support 'running' equipment.

Case Study 8

A complaint by a customer reported excessive vibration from an 11" diameter axial flow fan running at 3550 rpm on a 60 Hz A.C. supply. A visit to site showed that the customer was correct and it was arranged to take a full vibration trace, using a narrow band filter to assist in the analysis. Investigation of the trace showed that excessive vibration was being produced at 15 x rpm.

As the fan had 8 blades and 7 guide vanes, an aerodynamic source was suspected. The fan was measured and it was found that the clearance between the impeller blades and guide vanes increased diametrically to the extent of only 1/16". When the vanes were machined to remove this error, the problem was solved.

The correct measuring equipment and a knowledge of fan noise characteristics enabled quick identification of the problem.

Small dimensional "errors" may sometimes be the cause of a significant acoustic problem.

Case Study 9

The air supply to a ventilation system serving a new complex was provided by an air handling unit sited in a plant room immediately behind the screen of a basement cinema. Attenuators had been selected by the contractor, based on the sound power data for the air handling unit, to meet NC30 in the cinema and NC35 in a small hall on the floor above.

When the plant was first run there were no noise problems evident and sound measurements were not taken. However, the air flow to both areas was found to be low and the fan speed was increased from 1100 rpm to 1550 rpm. This caused a noise problem; the noise level in the cinema increased to nearly NC 45, made worse by a large amount of objectionable low frequency noise, Fig 10.1. The cinema, the small hall, and the corridor outside the cinema above which passed the main duct and attenuator in a false ceiling were affected.

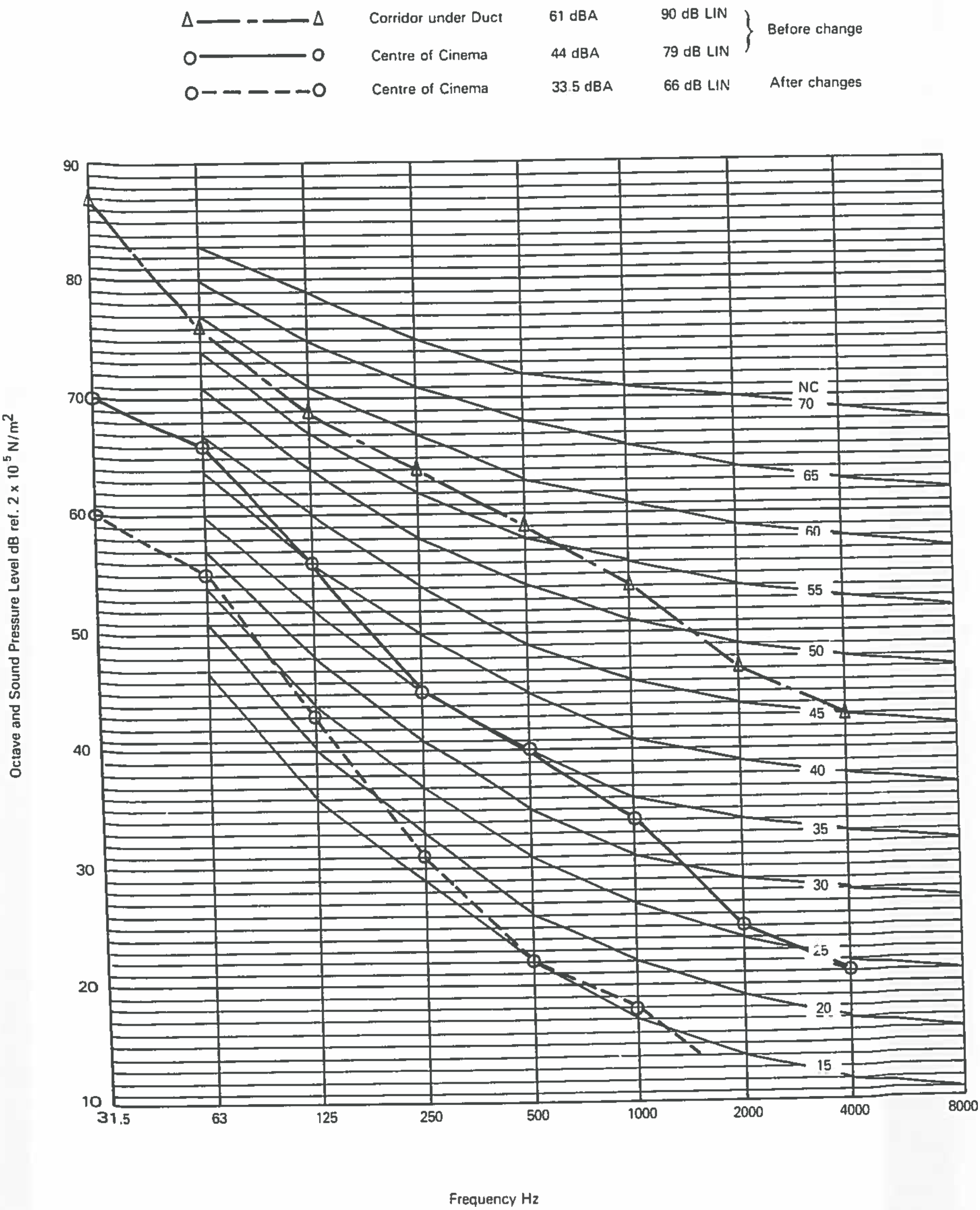
Narrow band analysis revealed random low frequency noise together with a large component at 26Hz — the fan shaft rotation frequency. This low frequency noise was reaching the cinema, small hall and corridor, along the ductwork, but the main path was via duct breakout before the attenuators.

The cause of the problem was clearly the very poor aerodynamic conditions between the air handling unit and the attenuator, Fig 10.2. Not only was the correct amount of air expected to divide itself (unequally) to serve the two parts of the system but the tortuous ductwork caused severe low frequency turbulence and, hence, noise. In addition, the 26Hz pure tone indicated an aerodynamic out-of-balance of the fan, probably caused directly or indirectly by the shaft "whirling" at the increased speed. To reduce the speed in order to improve the noise levels and maintain the required air quantity clearly required an improvement in the aerodynamics of the system, but there was no space to straighten out the ductwork. The problem was solved by inserting short chord turning vanes across each bend and across the 'plenum' of the fan discharge, see Figs 10.2. and 10.3.

When this was done, it was found possible to reduce the fan speed to just over 1000 rpm and still produce the required air flow.

The final noise level of just over NC25 in the cinema is shown on Fig 10.1. All the low frequency noise in the system was eliminated.

A low frequency noise problem arising, not out of an acoustic fault but an aerodynamic one. It was


Corridor under Duct 61 dBA 90 dB LIN
Centre of Cinema 44 dBA 79 dB LIN
Before change
Centre of Cinema 33.5 dBA 66 dB LIN After changes
Octave and Sound Pressure Level dB ref. 2 x 10 5 N/m2
90
80
70
60
50
40
30
20
10
31.5
63
125
250
500
1000
2000
4000
8000
NC
70
65
60
55
50
45
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35
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25
20
15
Frequency Hz

Fig. 10.1.

solved by improving the airflow conditions and not by the application of any noise control equipment.

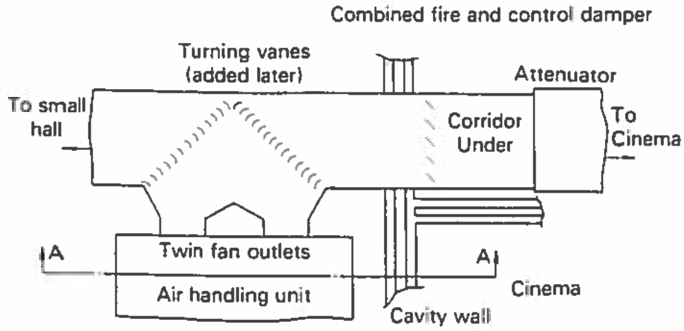


Fig. 10.2.

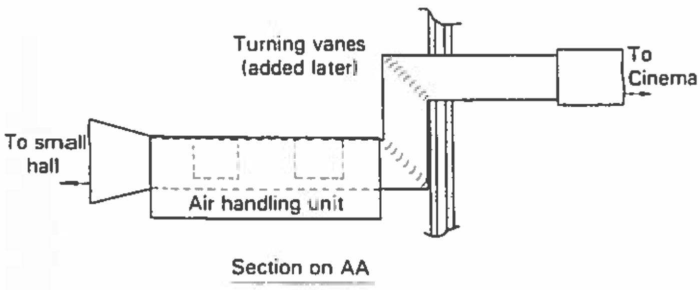


Fig. 10.3.

Case Study 10

When a Gas Cleaning Plant serving metal casting cupolas became fully operational 24 hours a day, complaints about noise began to arrive from the local neighbourhood. In certain cases not so local — it was claimed by some that the plant could be heard seven miles away!

Essentially the gases from the hoods above the cupolas are drawn through a long run of circular ductwork into a water-scrubbing unit powered by a large industrial centrifugal fan situated on the ground next to the scrubbing unit. The dirty exhaust gases pass upward through the water scrubber and after cleaning pass round a 180° bend and so down into the fan. The fan exhausts into a 26 metre high stack.

At the design stage a specification laid down noise levels not to be exceeded at the factory boundary fence and cylindrical pod attenuators were incorporated in the ductwork on both sides of the fan. The fan itself had been provided with a small enclosure. The selection of the pod attenuators had been based on octave band sound power level data provided by the fan manufacturer.

Noise measurements established that the problem was due to a pronounced tone at 271Hz, the fundamental blade passage frequency of the fan. (The fan has 11 blades and rotated at 1475 rpm.) The dominant noise source for transmission to large distances was radiation from the top of the stack. An additional larger enclosure had been constructed around the existing fan enclosure and the base of the stack. This had been effective close to the fan, but the noise remained unabated at distances greater than 100 metres from the plant. Fig 10.4.

Enquiries elicited the information that:

(1) It was thought that the two attenuators were severely clogged.
(2) A similar plant at another factory had no noise problems, and
(3) The fan was operating at the 'design point' on its characteristic — moving 35,000 cfm through 46ins W G — pressure drop at 77°C.

In-duct sound pressure measurements were taken above and below the pod attenuator in the stack using a 1 inch microphone and nose cone.

These confirmed that the attenuator was not effective. The Sound Power Levels calculated from the measured Sound Pressure Levels differed considerably from the data provided by the fan manufacturer. The most startling difference was in the 250Hz octave band — the octave band containing the 271Hz pure tone, where measurements indicated the level in the 250Hz band to be over 18dB above that in the 500Hz band whereas the fan manufacturers data indicated a difference of only 6dB. Readings showed a pressure drop of 62ins W G across the fan and a volume flow rate of 25,500 cfm. The fan was clearly throttled and examination of the plant showed that the scrubber itself was virtually clogged.

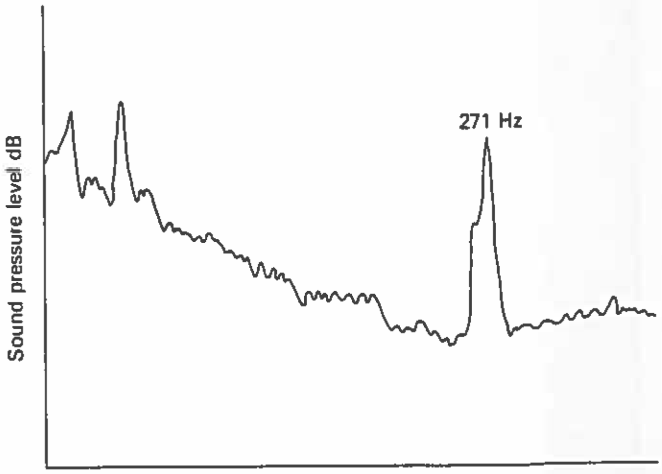


Fig. 10.4. Narrow band analysis of background noise at a residential location half a mile away from factory.

A return visit was made when the scrubber had been cleaned out and a repeat series of noise measurements were taken with the plant manual throttle device adjusted to give a series of pressure drops and volume flows. A clear correlation was evident, the more throttled the fan (i.e., the lower the volume flow and the higher the pressure drop) the higher the level of pure tone, the difference between the best and worst cases being 26dB in the 250Hz octave band level. Fig 10.5.

Investigation revealed that it was necessary to partially throttle the fan on start-up to avoid excessive current demand for the fan motor. It was then usual to leave the throttle in this position, the plant thus producing a high level of 271Hz pure tone.

The solution proposed was a dual one. The outlet attenuator was replaced by a more effective splitter type to cope with the short periods when the plant is run with the scrubber partially clogged and the fan throttling device was modified to ensure that the fan operates as near to its design point as is practical.

Subsequently it was discovered that an identical plant which had no noise problems has recently proved to be the source of an irritating noise!

An unusual application, but one which nevertheless illustrates the most important rule of site investigations — that ALL information, however helpfully intended, must be checked!

Technically, the case emphasises the importance of operating a fan as near to its design point as is possible.

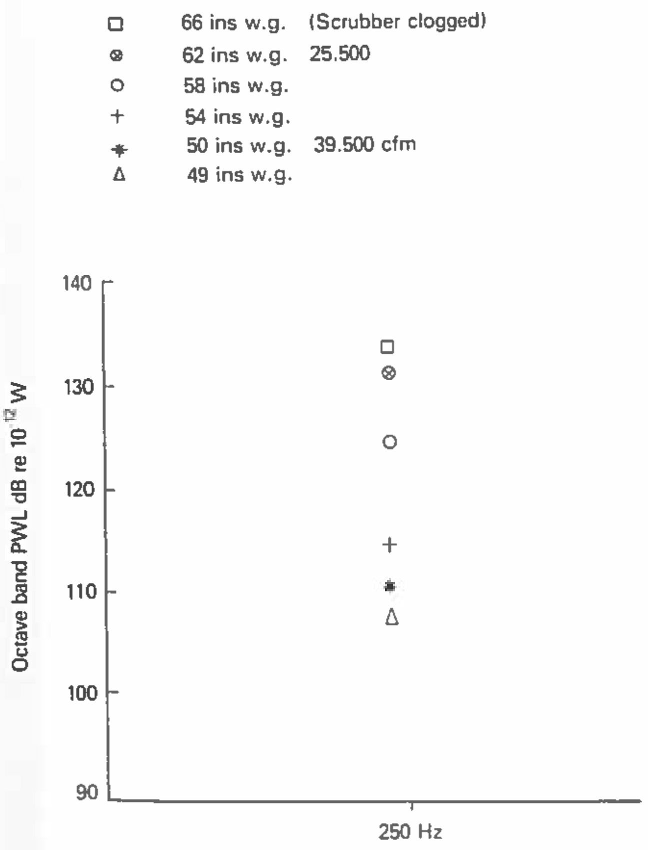


Fig. 10.5. Fan sound power levels at various operating conditions

Case Study 11

In a large auditorium, the supply air duct dropped from the plantroom into the conditioned space, and then entered the structure again, to supply into an underseat plenum system. When the system was commissioned it was found that the noise level was in excess of the design level of NC20. Investigations showed that the problem was mainly a low frequency one, being due to noise breakout from the large area of rectangular supply duct, which was approximately 10 metres long. The subjective effect of the noise was worse than the measured NC25 would suggest, as much of the low frequency noise was below the 63Hz octave band, i.e., less than 44Hz, and consequently did not show up on an NC chart, which starts on the 63Hz band.

Three options were available:

i. Reduce the noise at source.
ii. Provide more attenuation.
iii. Stop the noise breaking out of the duct.

With any acoustic problem, step i. is always to be preferred. Step iii. would have required some very heavy duct wall enclosure, and the only guaranteed solution would be to enclose it within a concrete duct, which was not possible for many reasons. Contractually, it was necessary to investigate the attenuator performance as this was considered to be less than the specification required. On the basis of site tests, further attenuation was installed in the plantroom, which marginally improved the situation.

However, the problem being of a very low frequency nature, normal absorptive attenuators are not effective, and any substantial reduction in noise level would have come from reducing the fan noise. The system volume was lower than design, a common source of excessive low frequency fan noise, and investigation of the fan noise data showed that the factory measured levels were higher than the quoted figures so that the manufacturers deemed it necessary to put an acoustic enclosure around the fan. It is still not clear what the real purpose of this enclosure was, as it would not alter the induct sound power level, and as it did not have a floor the plant room noise level was the same, if not higher than, without it. The enclosure was very close to the inlet eye of the fan, i.e., less than one diameter, and air was drawn in from the partially open top of the enclosure.

Because it seemed likely that the enclosure was giving rise to poor intake conditions to the fan it was dismantled. With the enclosure panels removed, the system volume increased and the low frequency noise reduced, to the point where it was considered acceptable. Breakout noise is still audible, but this is now a limitation of the normal duct breakout through the large areas of rectangular ductwork.

This case history shows that the following points must be considered in detail for low noise level areas.

1. *Fans should be selected for optimum efficiency, and site airflow tests conducted to ensure that they are working on the correct part of their characteristics.*
2. *Noise control equipment must not alter the airflow conditions into fans.*
3. *Main attenuators must be located in the plant room.*

4. *Fans should not discharge directly into conditioned areas.*
5. *Large areas of rectangular ductwork must be avoided within the conditioned space.*

Case Study 12

A number of warm air supply units were fitted in a new extension to a factory. The units comprised a cowl incorporating filters, a recirculation section, heater battery, fan and four-way discharge head.

During commissioning the units were run and it was said that they were much noisier than similar units fitted in the original building.

Measurements proved that the noise levels in the extension were much higher than the calculations, based on the building characteristics and the supply unit manufacturer's published data, would have led one to expect.

Investigation revealed the interesting fact that the manufacturer quoted the same noise spectrum for a given size and speed of fan irrespective of the type of installation. It was assumed that the catalogue showed the lowest possible spectra!

Knowing the dimensions of the fan and casing, a typical in-duct spectrum was assumed and a very approximate value for the actual noise output of the supply unit was determined. Using the revised figures to predict the noise levels in the extension gave a result reasonably close to the measured value.

Unfortunately it was not possible to suggest an acceptable modification since there was insufficient space to accept either a larger fan or silencer. However, when production commenced the manufacturing background noise substantially masked the unit noise and it was eventually agreed that the new supply units were no noisier than the acceptable models in the original factory.

This is a case where the fan was too noisy!

Check that the fan spectrum is the correct one for the application — if in doubt, ask the manufacturer.

Beware of subjective impressions formed when areas are unoccupied — they are usually wrong.

Case Study 13

In order to prevent too much noise being transmitted down a length of ducting a splitter silencer was fitted downstream of the fan. In spite of this precaution, complaints of excessive noise were received from the occupants of the offices served by the system. Measurements verified that the complaints were justified and the fan was blamed for being too noisy.

The fan manufacturer, who was confident of the acoustic characteristics of his product, looked elsewhere in the system for the source of the noise.

Two causes were found. The silencer was positioned too close to the fan discharge and the consequent turbulence produced noise. The average face velocity of the air entering the silencer was well above that recommended and the silencer itself was generating a considerable amount of noise.

A larger silencer was installed, and positioned further from the fan, with the result that the system was acceptable.

Always design sufficient space between components, especially at the fan outlet.

Do not guess a silencer size. Like any other component, it must be correctly selected.

Case Study 14

Two large double inlet backward curved centrifugal fans were incorporated in air handling units situated in roof top plant rooms supplying conditioned air to a multi-storey office block. The fans produced a very intense low frequency pure tone in the offices immediately below the plantroom. Apart from this pure tone, the noise level within the conditioned spaces was about NC35. The level of the pure tone varied, but typically raised the 31.5 octave band reading to 85dB.

At first the problem was thought to be due to a resonance in the system which supported the air handling unit. However, investigation showed that the pure tone did not correspond with the fan running speed of 23Hz but occurred at 33Hz; also if suitable combinations of access panels were removed, thus increasing the airflow through the fan, the pure tone disappeared. This led to the deduction that the problem was due to the fan stalling and reacting with the ductwork system to produce a resonance.

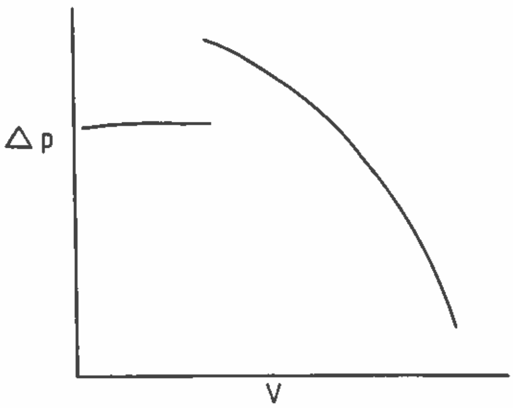
Backward curved fans are usually considered not to stall. This is because they do not exhibit the abrupt stall of a forward curved fan or a high pitch angle axial flow fan. Fig 10.6. When the volume flow through a forward curved fan falls below a critical value, the pressure rise drops sharply as all of the blades stall more or less simultaneously. With a backward curved fan, only one or two blades stall initially, and these cause adjacent ones to stall, while the originally stalled blades recover. This leads to a "cell" of stalled blades which move round the impeller. As the flow is further reduced more and more blades stall, but there are still other blades working more or less normally. This enables the fan to continue to produce its normal pressure rise as the flow rate is reduced, with varying numbers of blades working or stalled depending on the flow rate required.

Since the stalled area moves round the impeller which is itself rotating the fan is capable of generating aerodynamic noise at frequencies other than shaft speed. If there is a suitable duct element which is tuned near to the natural rate at which stall cells rotate, intense pure tones can be produced. The intense levels and low frequencies often produce 'trouser shaking' noise levels at the fan intake or discharge.

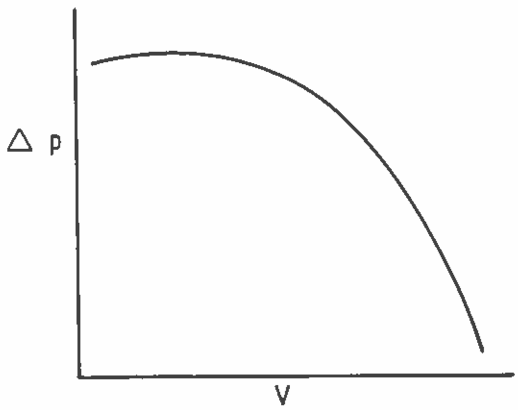
Obviously the system resistance was such that the fan was working on the stalled part of its characteristic. The initial fan selection was unwise — it would have been better to have a larger diameter single width fan which would have been capable of developing a high pressure at a lower volume flow. Fig 10.7.

The problem was cured by blanking off one half of each of the fans reducing them to single inlet units.

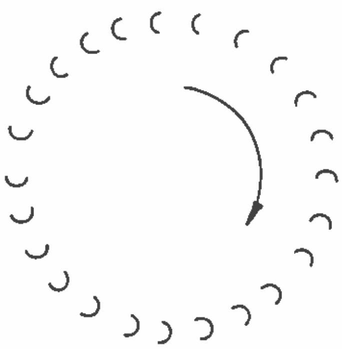
A most interesting example which emphasises the fact that fans, must never run on, or near, the stalled part of their characteristic.



Forward Curved Fan all blades stall simultaneously leading to collapse of flow.



Backward Curved Fan individual passages stall while the remainder function more or less normally.



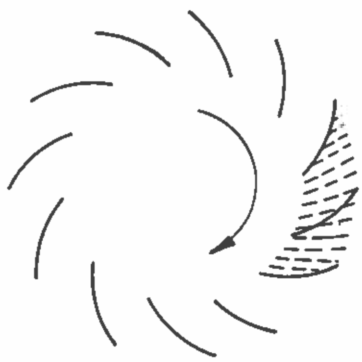
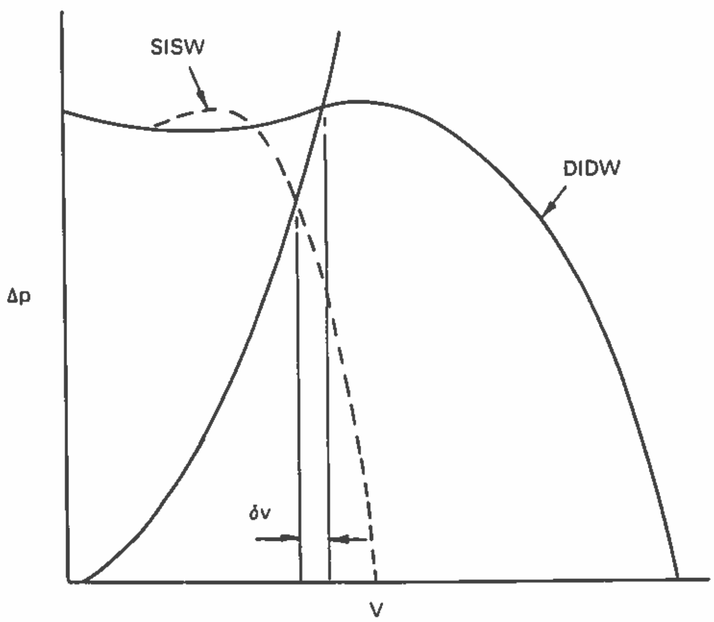


Fig. 10.6.



Changing to a SISW (Single inlet single width) fan of the same diameter as the original DIDW (Double inlet double width) fan enables system to operate in stable range with only a small reduction in flow rate.

Fig. 10.7.